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PRECISION MEASUREMENT OF GEAR LUBRICANT LOAD-CARRYING CAPACITY (FEASIBILITY STUDY)

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This investigation addresses the imprecision problem exhibited by the Ryder Gear Test (ASTM D-1947) presently utilized as a gear lubricant load-carrying capacity qualification test under such specifications as MIL-L-7808 and MIL-L-23699. The primary program objective is to determine the feasibility of developing a precision technique for measuring gear lubricant load-carrying capacity suitable to replace the current costly and imprecise Ryder Gear Test.

Program efforts involved the thorough analysis of the Ryder precision problem as well as the three replacement technique alternatives: modified Ryder, existing precision technique, and development of a new precision technique. Results of these analyses indicated the need for development of both a new load-carrying capacity rating criteria as well as a new test machine.

The revised criteria that has been proposed involves a three step characterization process which includes wear, scoring load, and post scoring recovery time. This criteria will be implemented through a real time monitoring approach.

The proposed precision load-carrying capacity test device is a disc on disc approach. Critical precision parameters include continuous loading, test specimen cooling, precision control of specimen quality, and precision control of operating parameters.

Utilization of the new criteria in conjunction with the advanced disc on disc machine will provide a cost effective, precision determination of gear lubricant load-carrying capacity. Primary program recommendations include the design, fabrication, and testing of the proposed new test approach.

FOREWORD

This report presents the results of a study conducted by Mechanical Technology Incorporated (MTI) for the Aero Propulsion Laboratory, Air Force Systems Command, Wright Patterson Air Force Base, Ohio, under contract F33615-80-C-2016. The study was based upon investigations carried out to explore the feasibility of developing a precision technique for measuring lubricant gear load-carrying capacity to replace the Ryder Gear Test (ASTM D-1947).

The work was performed under the direction of Mr. Leon DeBrohun, Air Force Project Engineer. Mr. Peter Senholzi was the Program Manager for this contract at MTI, with Mr. N. Suresh Rao providing the technical and analytical direction. Mr. Alan Maciejewski served as Project Engineer and Technical Analysis Coordinator.

The authors wish to acknowledge the program assistance provided by Mr. Leon DeBrohun and other individuals of the Air Force. Appreciation is also extended to Mr. P. Mangione of the Naval Air Propulsion Center for his contributions to this study.

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EXECUTIVE SUMMARY

This investigation addresses the imprecision problem exhibited by the Ryder Gear Test (ASTM D-1947) presently utilized as a gear lubricant load-carrying capacity qualification test under such specifications as MIL-L-7808 and MIL-L-23699. The primary program objective is to determine the feasibility of developing a precision technique for measuring gear lubricant load-carrying capacity suitable to replace the current costly and imprecise Ryder Gear Test.

Program efforts involved the thorough analysis of the Ryder precision problem as well as the three replacement technique alternatives: modified Ryder, existing precision technique, and development of a new precision technique. Results of these analyses indicated the need for development of both a new load-carrying capacity rating criteria as well as a new test machine.

The revised criteria that has been proposed involves a three step characterization process which includes wear, scoring load, and post scoring recovery time. This criteria will be implemented through a real time monitoring approach.

The proposed precision load-carrying capacity test device is a disc on disc approach. Critical precision parameters include continuous loading, test specimen cooling, precision control of specimen quality, and precision control of operating parameters.

Utilization of the new criteria in conjunction with the advanced disc on disc machine will provide a cost effective, precision determination of gear lubricant load-carrying capacity. Primary program recommendations include the design, fabrication, and testing of the proposed new test approach.

1.0 INTRODUCTION

Mechanical system reliability and durability are a function of both structural integrity and wear integrity. Emphasis to date has been placed on structural integrity with a "throw away" philosophy accommodating the consequences of wear integrity. Recent resource limitations, however, have promoted substantial interest into the area of wear integrity optimization. The optimization process is approached from the aspects of wear prevention and wear control. Wear prevention occurs primarily in the equipment design process, while wear control is instituted in the operational arena.

The process of wear integrity optimization involves addressing the variables of wear either individually and/or in combination both in the design and operation of mechanical components/systems. Such variables as materials, surfaces, lubricants, additives, and contaminants must be considered with respect to design constraints, operating parameters, and operating environments.

A key element in the wear integrity optimization process is the existance of viable wear test techniques or tribo-testing techniques. These test techniques are utilized in the research, development, qualification, quality assurance, and troubleshooting arenas.

- Research - Testing is utilized in the research arena to study fundamental wear mechanisms.

- Equipment Development - Testing is utilized in the development process to define wear variables and verify overall design wear integrity.

- Qualification/Quality Assurance - Under the qualification arena, testing is utilized to both qualify and verify conformance to a design specification.

- Troubleshooting - Testing is utilized in the troubleshooting arena to upgrade wear variables in order to accommodate a misapplication or a change in application.

Wear testing includes field, system, component, and simulation testing. As a result of time and cost constraints, accelerated simulation testing is emphasized over component, systems, and field testing. As reported by P. B. Senholzi in "European Tribology Technology: An Assessment Of The State-Of-The-Art," (ONR London, July 1978), there exists both in the United States and Europe extensive proliferation of simulation techniques. These techniques overall exhibit poor repeatability within a test facility, poor reproducibility between test facilities, and poor correlation with actual field performance. As a result of these characteristics, simulation test techniques are primarily utilized as a wear variable ranking tool as opposed to a quantitative variable assessment tool.

Lubricant performance testing is a major facet of the tribo-testing arena. Test requirements include lubricant development, qualification, quality assurances, and troubleshooting. Lubricant performance testing includes finite, component, system, and field testing levels. As discussed above, there currently exists extensive proliferation of test techniques, approaches, and sequences. These numerous test alternatives vary considerably with respect to time, cost, repeatability, precision, and field correlation.

One such test technique, as discussed above, is the qualification tests for gear lubricant load-carrying ability. The ASTM Standard Test Method D-1947 "Load-Carrying Capacity of Petroleum Oil and Synthetic Fluid Gear Lubricants," describes the test apparatus and procedure required by the technique. As a result of the relatively poor repeatability and reproducibility of this technique, it is necessary to run numerous tests in order to establish test result confidence/significance. This required test approach results in substantial personnel, time, material, and thus cost expenditures.

The program described in the following sections of this document is aimed at the feasibility determination of developing a precision technique for measuring gear lubricant load-carrying capacity. This precision technique would replace the current Ryder Gear Test as performed under the ASTM Standard Test Method D-1947.

2.0 GENERAL DISCUSSION

The ASTM D-1947 test,generally referred to as the Ryder Gear Test, is a standard test for measuring the load-carrying capacity (L.C.C.) of petroleum oil and synthetic fluid gear lubricants (1). This test is prescribed for the qualification of aviation synthetic lubricants such as MIL-L-7808H and MIL-L-23699C. For quite some time, a controversy has existed with respect to the total Ryder test program (i.e. hardware, calibration fluid, and technique used for reporting L.C.C.) (2). Inconsistencies have been noticed in the mean L.C.C. ratings of reference oil C which is used as a calibration fluid for checking the extent of standardization of the hardware/device used and as a check to ensure compliance with the prescribed test technique and test procedure. These inconsistencies have been observed between different batches of standard test gears purchased from a common source; between the three test heads - Ryder, WADD, EAF, all of which have been approved for use for the above test; and between approved participating laboratories from year to year and sometimes within the same calendar year with one test head and device.

2.1 Reference Oil C Program

As part of a continuing program under an Air Force grant, the Southwest Research Institute (SWRI) issues the Reference Oil C status reports on a yearly basis by compiling the L.C.C. data obtained from participating laboratories (3). The Reference Oil C (a batch of 200 drums retained by SWRI, obtained from Humble Oil and Refining Company, to specification MIL-L-6082C, Grade 1100 engine oil) was approved by AFAPL in 1965 as a replacement to Reference Oil B (a batch of 200 drums manufactured in 1957 by Humble Oil and Refining Company, to specification MIL-L-6082B, Grade 1100). A scrutiny of these status reports issued between 1965 and 1971 confirms the above inconsistencies. Generally speaking, the results of the Reference Oil C program indicates that the load-carrying capacity of the reference oil may, in general, be influenced by: the test head type (Ryder, WADD, EAF), the test laboratory, and the vintage/batch nos. of the standard test gears prescribed for the test.

2.2 CRC-Program on the ASTM D-1947 Test (1971-1974)

In an effort to determine the reason for this behavior, the Coordinating Research Council (CRC) - jointly sponsored by API and SAE with a broad based membership in the automotive fuels and lubricant testing industry including U. S. Army, Navy, and Air Force - established a research project. The above CRC program was conducted during 1971-1974 with one test device by one laboratory (Alcor Inc.) which performed the tests under contract to the CRC, using an ERDCO ANTI-FRICTION (EAF) test head, a WRIGHT AIR DEVELOPMENT DIVISION (WADD) test head, and a RYDER test head. To accomplish the program, the contractor used two test fluids and two groups of Ryder test gears. One group of gears was of relatively earlier manufacture than the other. The results of this program, concluded with a total of 48 test gears, confirmed the existance of a precision problem with the ASTM D-1947 test. Although specific reasons for the lack of precision were not identified, it was generally confirmed that the vintage of test gears, the design of test heads, and the reference fluid all had varying degrees of influence on the precision of the test. Test gears of an earlier vintage, generally speaking, gave lower values of L.C.C. ratings to the test fluids (i.e. Reference Oil C and another synthetic fluid - Hercolube A). The average load-carrying capacity of Reference Oil C rated at 3046 lb/in. as compared to the synthetic oil (Hercolube A) which rated at 2051 lb/in. Since completion of the above program, strong recommendations were made in favor of alteration of both the test method and test hardware. To reach the above objectives, it was recommended that studies be carried out to establish the influence and relationship of specific test gear parameters on the load-carrying capacity and to study the lubricant/metallurgy interaction effects on lubricant load-carrying capacity rating (2).

2.3 SWRI-Study (1973-1976)

As a sequel to the CRC program, studies sponsored by the U. S. Air Force were conducted at the Southwest Research Institute (SWRI) during 1973-1976, to address the above problems from both a theoretical as well as an experimental approach (4). In this study Boundary, EHD, and Critical Temperature concepts of Archard, Dowson, and Blok were critically reviewed and the influence of some of the operational and system parameters likely to affect precision in the determination of load-carrying capacity were investigated from theoretical considerations (4). Under the experimental approach, the gear scuffing program carried out, comprized of 64 determinations at 4 different speeds - 2500, 5000, 10,000, and 15,000 r.p.m. and by two different scuffing load characterization criteria. The failure loads were recorded for both 10% area scuffing of the test gears as well as 22.5% area scuffing of the gears. The oils tested were of the MIL-L-7808 type and the WADD No. 2 test machine was used in this study. From the results obtained in this program, SWRI has noted that "there is generally no consistent relationship between the values of scuff-limited load-carrying capacity for any individual test; However, taken as a whole, that is, comparing the mean values, the 10% (scuff) values are about 10% lower than the 22.5% (scuff) values." For the first time, the rating criterion for characterizing results was identified as a new parameter influencing test precision in addition to test machine type, test gears and their vintage, test laboratory and test oil. Also, the SWRI results indicated that the higher the test speed the lower appeared to be the scuff related failure load (4).

In the eleven year history of the ASTM D-1947 test, traced briefly under sections 2.1, 2.2, 2.3, it is noticed that the precision problem is evident when reproducibility of results between laboratories and test heads are considered and also when the results within one laboratory and one test device are considered. These investigations, however, have not revealed/pin-pointed specific parameters affecting the said precision problem beyond generic references to test machine head employed (Ryder, WADD, EAF), vintage of test gears, laboratory performing tests, and reference oil used (B or C). The theoretical analysis carried out in the SWRI study under spur gear mechanics indicates the complex nature of the influence of test head design features such as dynamic loading factors, misalignment caused between mating gears due to misalignment of support bearings, elastic deflections and differential thermal expansion of shafts, support bearings and housing, not to mention the influence of tooth errors (pitch, deflection, and profile) in test gears and other auxiliary transmission gears. It seems impossible to investigate the effect of each of these design features on the load-carrying capacity rating. The important system and operating variables affecting precision will be separately dealt with under Section 4.2.2 of this report.

The reality of the precision problem outlined with the ASTM D-1947 test has prompted the U. S. Air Force to look for alternate test techniques to replace the Ryder Gear Test, at least from the point of view of qualification of aviation synthetic oils. In this regard, Mechanical Technology Incorporated (MTI) has undertaken the present feasibility study on precision measurement of lubricant gear load, under Contract No. F33615-80-C-2016 under the sponsorship of Wright-Patterson Air Force Base, Ohio.

3.0 ANALYTICAL PROGRAM APPROACH

The general review of the Ryder precision problem, presented under Section 2.0, and guidelines laid for the present feasibility study (Contract No. F33615-80-C-2016), led to the development of the following MTI program approach.

PHASE I, SURVEY

Survey of pertinent literature in order to accomplish the following objectives:

- Terminology development by review of the present understanding of Boundary, EHD, and gear lubrication concepts.
- Gear failure modes - discussion
- Listing of available information on the Ryder gear precision problem and on variables affecting the Ryder Gear Test.
- Listing of potential Ryder Gear Test modifications and discussion of alternative characterization concepts.
- Listing of existing alternative gear test machines and other film strength test techniques/apparatus.

PHASE II, ANALYSIS OF SURVEY

- Analysis of existing alternative test techniques and their respective evaluation criteria.
- Analysis of criteria for characterizing load-carrying capacity in the Boundary lubrication regime.
- Analysis of parameters influencing load-carrying capacity determination by film strength testing in general.

PHASE III, SELECTION

- Development of criteria and weighing factors for the selection of an alternate technique to the Ryder Gear Test.

- Selection of the alternate test.

- Explanation of the selection process.

- Comparison of the selected alternative with existing similar techniques.

The results of each of these work phases will be discussed and outlined in the following Sections; 4.0, 5.0, and 6.0, respectively.

4.0 PHASE I, SURVEY

The survey approach was comprised of computer based literature searches, personal contacts, and review of on-going programs. Over 100 technical papers and reports were gathered in the process and reviewed. Drawings, specifications of the Ryder Test Gears, MIL specifications, and information on test costs and hardware costs were obtained through contacts and direct correspondence with concerned people.

4.1 Terminology and Concepts

The scope of Phase I (Survey) includes a clearer understanding of the interdependence of the meshing action of gears and their modes of failure with lubrication concepts. The purpose of this effort is to make the analysis of the maze of system, operational and material parameters influencing load-carrying capacity determinations easier. In order to accomplish this, the scope of Phase I was slightly broadened to include discussions/reviews of the current thinking on topics such as: meshing action of gears and film formation; influence of geometry and mechanics on film formation and failure; discussions on gear lubrication with emphasis on Boundary and EHD lubrication concepts; and the role of lubricants and lubricant additives used in aircraft turbine oils particularly to improve film strength and to meet qualification requirements in load-carrying capacity. This section on terminology and concepts will be discussed as follows:

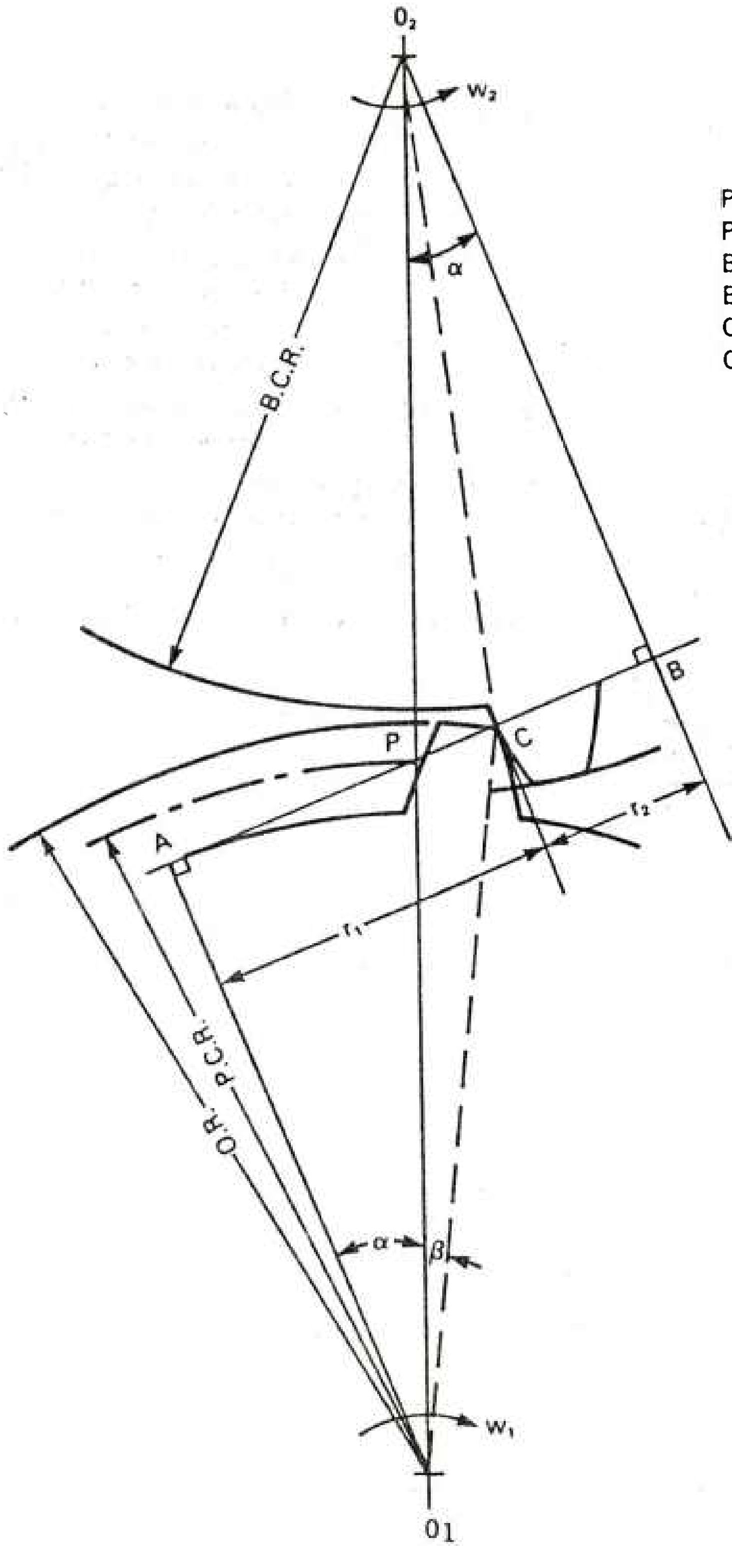
- Gear Motion and Meshing Action
- Gear Failure Modes
- Gear Lubrication
- Gear Lubricant Additives
- Lubricant Testing

4.1.1 Gear Motion and Meshing Action

Gear Motion as explained by Balmforth, gives a quick understanding of the concepts involved (5). These concepts have been generously used in this section. Gear contacts can be broadly divided into two classes. The first includes spur, helical, straight, and spiral bevel, and the second, worm and hypoid gears. In both groups, the contact is a combination of sliding and rolling, but the ratio of sliding to rolling is much greater in the second group than in the first. This difference has a great influence on the characteristics which are required in a suitable lubricant.

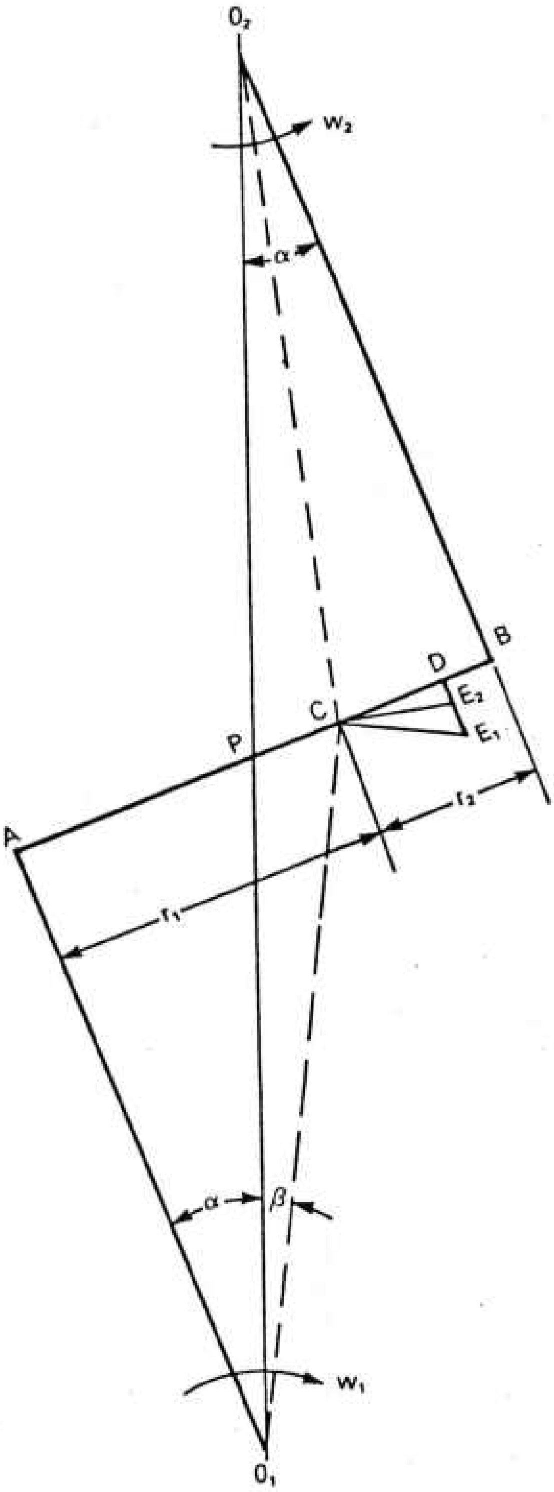
Consider the Ryder gear-mesh configuration (Figure 4-1), as an example of the first group. The path of contact lies on the common tangent to the base circles and the length of the contact path is limited by the outside diameter of the gears. The tooth action results in a combination of rolling and sliding between engaging profiles. Pure rolling occurs only at the pitch point and maximum sliding velocity at the tip of a tooth, point C, indicated in Figure 4-1.

The point of contact travels along the common tangent to the base circles and the common tangent is normal to the tooth profiles at the point of contact. Each of the contact points on mating flanks has a tangential velocity. The velocity vector diagram for the maximum sliding velocity at point C has been indicated in Figure 4-2. The maximum sliding velocity (V_s) across the line of action is the difference of the velocity components for point C (gear tip) across the line of action. The slide/roll ratio for point C is the ratio of sliding and rolling velocities at point C. At the point of contact, each tooth flank may be considered to be part of a cylinder, and therefore can be represented by two cylinders. This concept is important in representing gear contact motion by a pair of discs, by simulating the sliding velocity and slide/roll ratio at the desired point of contact.



P.C.D. = 3.5 in. (pitch circle dia.)
P.C.R. = 1.75 in.
B.C.D. = 3.2336 (base circle dia.)
B.C.R. = 1.6168 in.
O.D. = 3.72 in. (Outer Dia.)
O.R. = 1.86 in.
α = pressure angle = 22½°
$w_1 = w_2 =$ 10,000 r.p.m.

Figure 1. Ryder Gear-Mesh Geometry Diagram



$U_1 = \overline{CE_1} =$ Velocity vector for point C from driving gear.

$U_2 = \overline{CE_2} =$ Velocity vector for point C from driven gear.

$\overline{CD} =$ Velocity component of U_1, U_2 along line of action.

$V_1 = \overline{DE_1} =$ Velocity component of U_1 across line of action

$V_2 = \overline{DE_2} =$ Velocity component of U_2 across line of action

$V_s =$ Sliding velocity across line of action (Max) $= V_1 - V_2$

$V_s = \overline{DE_1} - \overline{DE_2}$

Slide/roll ratio (Max) $= V_s/(V_1 + V_2)$

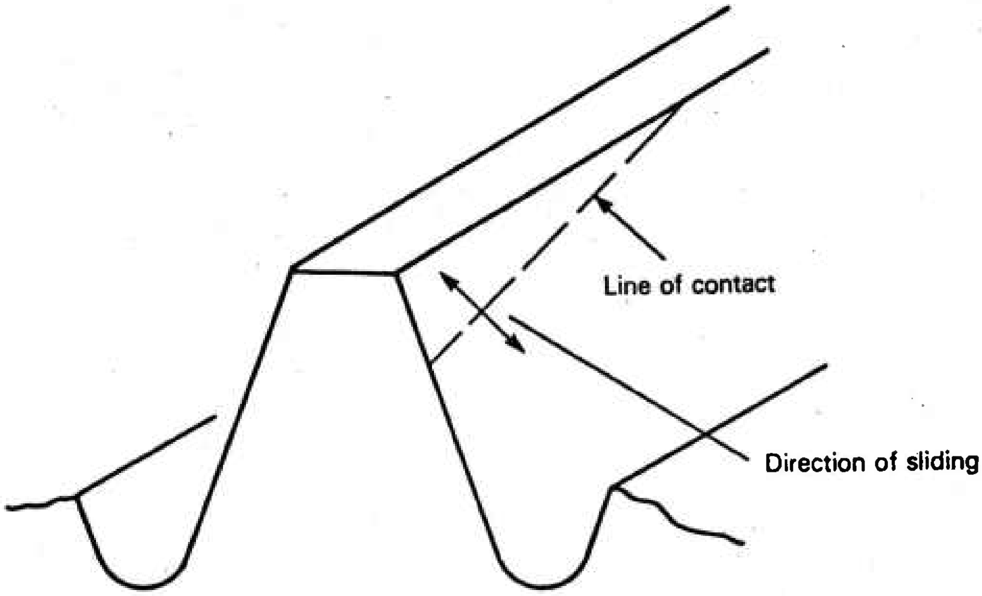
Figure 2. Ryder Gear-Mesh, Velocity Vector Diagram

If the contact is pure rolling, then conditions for the formation of a hydrodynamic film are good; friction is at a minimum and there is no shearing of the oil film. Where there is sliding, the tendency to draw oil into the wedge is less than in pure rolling. If the film formed is not sufficient to completely separate the surfaces, then metal to metal contact will occur and damage will be caused by the relative sliding. This is the main reason why scoring first manifests itself at the gear tip and generally in addendum area of gear tooth due to the relatively high sliding involved across the line of contact as noted in Figure 4-3. Even with pure rolling (i.e. pitch point), the film may not be capable of preventing metal to metal contact, and while damage can occur, this would usually be much less serious than that caused by sliding (5).

Referring to the second gear group, worm and hypoid gears, the tendency towards film formation is not quite so pronounced. Consider the case of worm gears as in Figure 4-4. This figure shows a series of lines of contact between the worm and wheel. On each line of contact, the approximate direction of sliding is shown by a number of points due to worm rotation. Even at those points where the direction of this sliding and the contact line subtend the largest angle, the direction of sliding is more parallel to the contact line than normal to it. This is the major difference in conditions of contact between the two classes of gears described. If the tendency to form a wedge is small, then it follows that if satisfactory lubrication is to be obtained, any lubricant applied must resist the wiping action which occurs in the case of worm and hypoid gears (5).

4.1.1.1 Influence of Geometry and Mechanics

Gear mechanics and geometry have considerable influence on the formation of lubricant films. While the role of sliding and rolling motions were considered under 4.1.1, the role of some of the design



(Reference 5)

Figure 3. Direction of Sliding Spur, Bevel, Helical Gears

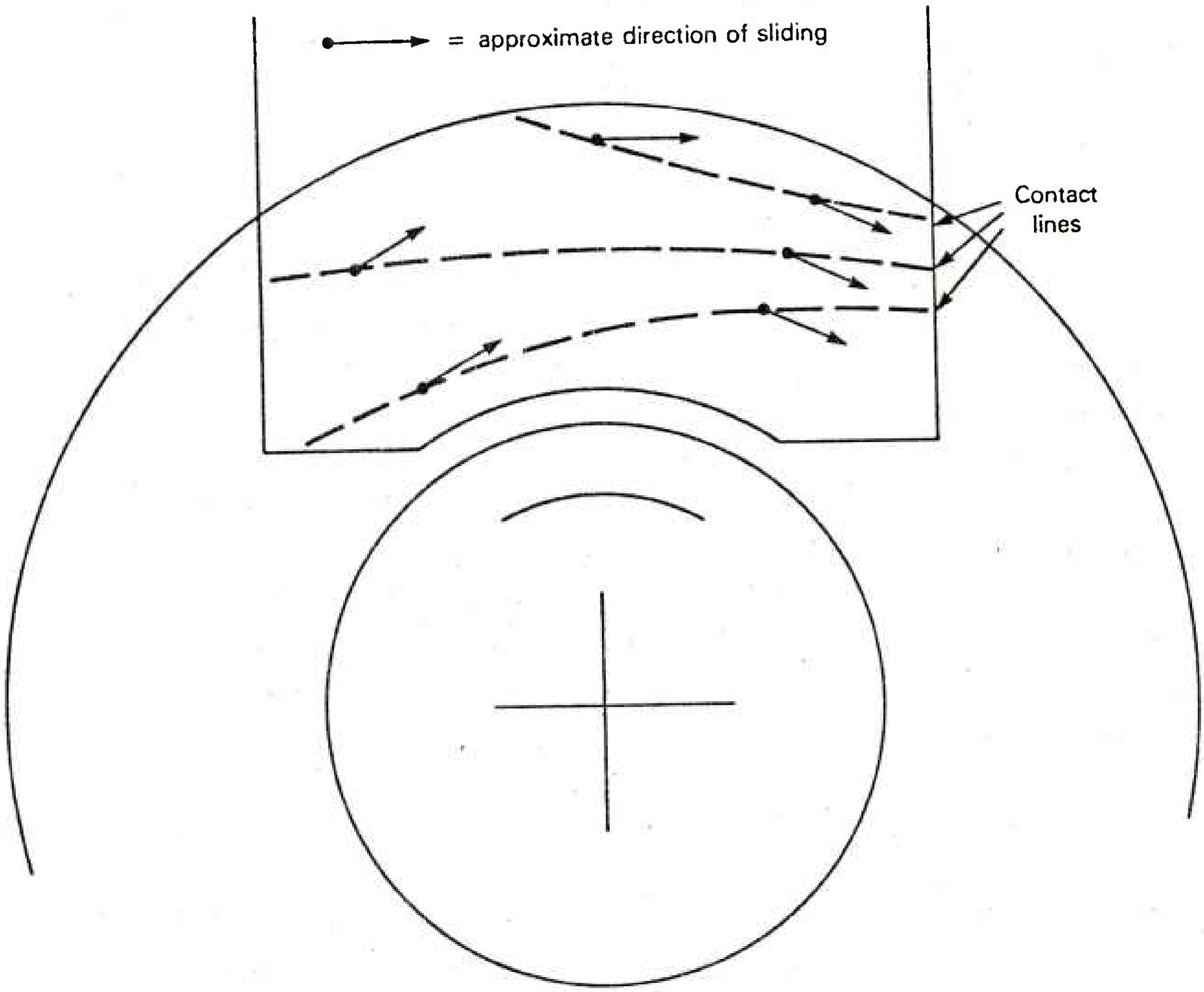


Figure 4. Worm Gear Contact

features and alignment of the meshing gears will be discussed here. In this respect the treatment given by P.M.Ku in Reference 4 is quite adequate and will be generously cited with appropriate changes in numbers referring to the literature cited thereof.

4.1.1.2 Geometry

Gears employ counterformal surfaces and are thus subject to high contact stresses. They experience relatively high sum velocities combined with relatively high sliding velocities, which may be cyclic or sustained depending on the gear type (4).

Gear kinematics can be precisely defined by assuming completely rigid gears (6-8). Even then, the subject acquires much complexity with such gear types as the hypoid, spiral bevel, and worm. In reality, gears are never completely rigid, hence one must deal with the interactions between forces acting and consequent surface deformation and tooth deflection (4).

4.1.1.3 Surface Deformation

Since gears are not completely rigid, one must consider the consequences of this fact. One important consequence is the local elastic deformation of the counterformal surfaces under load, which gives rise to elastohydrodynamic lubrication (4). The elastohydrodynamic lubrication concept will be separately explained under Section 4.1.3.1.

4.1.1.4 Tooth Deflection

The elastic deflection of the gear tooth necessitates tooth profile modification, which affects load sharing (9-11). Consider, for example, a set of involute spur gears (assuming no manufacturing errors) with a contact ratio* of less than two, for which the load is carried by two pairs of teeth at the beginning and end of the mesh cycle, and by only one pair of teeth during the remaining portion of the mesh cycle. In this simple case, the relation between the load sharing pattern and tooth profile modification for a particular design load can be established by statics with relative ease, but still with some measure of empiricism. If the contact ratio is, say, between two and three, the load is carried by three pairs of teeth at the beginning, middle, and end of the mesh cycle, and by two pairs of teeth during the remaining portions of the mesh cycle. The load sharing and profile modification problem of high contact ratio gears is considerably more difficult to solve. Design optimization is far more complex, because the propensity of both strength related and lubrication related failures depends markedly on how the high contact ratio is achieved (12). Nevertheless, high contact ratio normally exists in such gears as helical, hypoid, and worm; and it is gaining in popularity for aircraft spur gears.

4.1.1.5 Other Deflections

The gear bodies, shafts, support bearings, and housing also deform under load. These deflections may modify load sharing among the teeth, or cause tooth misalignment. Analysis of these deflections is even more difficult than that of tooth deflection; a rational approach is currently lacking.

4.1.1.6 Tooth Misalignment

Tooth misalignment may be due to the numerous bulk deflections mentioned above, manufacturing errors, stackup of tolerances in the assembly process, or differential thermal expansion. Whatever the causes, misalignment can substantially affect both strength related and

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* Note - Contact ratio is the ratio of the arc of action to the circular pitch. It is sometimes thought of as the average number of teeth in contact. For involute gears, the contact ratio is obtained most directly as the ratio of the length of action to the base pitch.

lubrication related failures (13-15). Misalignment is one of the more severe problems to handle, because it is difficult to measure and control in practice, and reliable prediction of its effects is still not available.

4.1.1.7 Dynamics

The dynamics of gear tooth behavior, due to the transient nature of tooth engagement, operation away from the profile modified design point, manufacturing errors, and externally imposed dynamic conditions, is an exceedingly complex subject. Clearly, if the actual tooth load is much higher than that derived for the static case, then estimates for both strength related and lubrication related failures based on the static load can be overly optimistic. The dynamics of gear teeth of simple geometry, under idealized conditions, has been a subject of much study (16-27), mainly with regard to strength related failures. Even so, the dynamics of a complete gear system, and also the dynamics of lubricant flow to and over the gear teeth, are quite different matters. The effects of gear dynamics and lubricant flow dynamics on lubrication related failures, as well as the time dependent chemical interactions involved in the failure processes, are by and large, not well understood at present.

4.1.2 Gear Failure Modes

As indicated by P. M. Ku and others, it is more practical to classify the gear tooth failure modes under two basic categories, namely, lubrication related modes and material strength related modes, although AGMA cites 21 modes of gear tooth failure (28) (4). Lubrication related failures include rubbing wear, and scoring/scuffing. Pitting mode falls under the second category of strength related modes along with plastic flow and tooth breakage. Because, wear (adhesive, abrasive) and scoring/scuffing are considered to occur when the normal oil film between teeth breaks down for some reason, surface fatigue failures (on the other hand) can occur even with proper lubrication and an unbroken oil film. As the term implies, they result from repeated stressing of the gear surface material, causing a crack to form on or near the surface. The crack gets larger with time until a small piece of gear material is removed. Fatigue failures can be detected only after an extended operating period of perhaps several million revolutions (with the exception of initial pitting due to high spots). This contrasts to scoring failure, which can become apparent almost immediately, and with wear failure which occurs immediately but may take some time before it becomes noticeable. The fatigue surface failure is known as pitting, which aptly describes the general appearance of the surface." (29).

These two basic gear failure mode categories will be elucidated separately in the following paragraphs.

4.1.2.1 Lubrication Related Failure Modes

The failure modes to be discussed in this category are wear (adhesive, abrasive, and corrosive), scoring, and scuffing of gears. A section on Blok's critical temperature hypothesis has been included in this section as this hypothesis is considered to offer the best rationale to explain scoring and scuffing modes of surface distress.

4.1.2.1.1 Wear

Wear is defined as removal of material due to mechanical or chemical action or both. Wear mechanisms may be initiated in three forms: adhesive wear, abrasive wear and corrosive wear.

Adhesive Wear is due to adhesion and breaking or transfer of contacting asperities when two surfaces rub together in the presence or absence of a lubricant. In the presence of lubricants, adhesive wear occurs when the film formed is insufficient to separate the surfaces. With lubricated surfaces, Abrasive Wear is due to the presence of harder foreign particles such as sand, oxides, carbon, etc., which get interposed between rubbing surfaces. With dry surfaces, abrasive wear is due to the shear difference in hardness of the mating surfaces, or due to foreign particles mentioned above. Sometimes abrasive wear is caused by build-up and work hardening of the debris from the surfaces themselves. Corrosive Wear is due to the attack of acids, moisture or other chemicals. Formation of such corrosive acids/chemicals could be due to the deterioration of the lubricant, the corrosivity of the lubricant itself, or due to contamination from combustion products or corrosive fuels especially in I. C. engines. Adhesive and abrasive wear modes have often been grouped under a single category, that of Rubbing Wear (4).

4.1.2.1.2 Scoring

Scoring is the sudden appearance of rough blackened spots at isolated locations across the striations or grinding marks of surfaces, giving the evidence of localized melting of asperities and their subsequent welding and tearing, probably due to intense heating of the contacting asperities under boundary lubrication conditions. Different viewpoints have been expressed on the mechanism of formation of scoring marks in gears and similar contraformal contacts. Moreover, scoring as defined in the USA is identical to scuffing as defined in the U.K. (30). Scoring is a symptom of inadequate load-carrying capacity of the lubricant or of overload of the teeth, much the same as wear. The appearance is that of a surface which has been welded to its mating surface and then torn loose, leaving a rough or matte finish. This is in contrast to the smooth grooves or polish of a worn surface. The tips and roots of the teeth are affected most, while the pitch line area is generally in its original condition. When gear alignment is correct and scoring is not due to isolated high spots on the tooth surfaces, the scored area will extend all the way across the width of the teeth (29).

4.1.2.1.3 Scuffing

Scuffing is identified as the growth or spread of rough spots from localized areas of a surface to larger areas of the surfaces, giving the appearance of metal having been melted and then roughed in subsequent rubbings. Scuffing is considered accumulated scoring, in other words, an advanced form of scoring. It could be a damage which appears as though a previously scored spot gets enlarged in size or a damage where new scored spots develop in and around the initial scoring mark extending the scoring damage to larger areas of the surface. Extensive scuffing damage can drastically alter the material properties (i.e. hardness and thermal conductivity) as also surface texture properties (i.e. roughness) of the interacting surfaces. It is therefore believed that scoring and its aggrevated form, scuffing, are the most important lubrication related distress modes to be tackled by proper lubrication.

Although the basic mechanism of the scuffing phenomenon is still largely not understood, there is good agreement that the breakdown of the EHD film is a necessary but insufficient condition for scuffing (4), (13), (31-45). In other words, in order for scuffing to occur, the operation must move not only into the boundary lubrication regime, but must also meet an additional requirement. However, largely because the mechanism of scuffing is basically unsettled, what form this additional scuffing criterion must take is still very much an open question. All available evidence appears to suggest that how deeply the operation may safely extend into the boundary lubrication regime without resulting in scuffing depends upon the physical and chemical nature of the oil, the metal and surface, the surrounding atmosphere, as well as the operating conditions. And if there is a generalized scuffing criterion, the concensus is that it is thermal in character, i.e., it is the consequence of the intense frictional heat generation at the potential failure site (4).

It should be remembered that the applied load is supported by both the area of metal to metal contact represented by contacting asperities as well as lubricant film separating the asperities at the valleys under boundary conditions. It is speculated that the thin lubricant films at the valleys would be carrying a major portion of the load till scoring/scuffing manifests and at this point of transition, where the lubricant films no longer carry/bear the major portion of the load, the burden of carrying the entire load suddenly falls on the contacting asperities. As a consequence, they are stressed beyond the yield point of the metal leading to localized melting and seizure with subsequent welding and tearing. The thermal scuffing model developed by Blok fully explains the flash temperature which suddenly increases the magnitude of the critical temperature required for failure of the thin films separating the surfaces (36, 37).

This discussion of scuffing as a failure mode would be incomplete without reviewing the critical temperature concept after Blok as this thermal model is considered to have the best rationale to explain scoring/scuffing. The Blok's critical temperature hypothesis will be discussed in the next section.

4.1.2.1.4 Critical Temperature Hypothesis

The following explanation of the Blok's concept has been derived from Reference (46). When two bodies are engaged in relative motion, heat is generated as a result of friction at the conjunction. When the bodies are gears or discs with all points on the surface repeatedly passing through a conjunction, the heat dissipation causes the surface temperature, as well as the temperatures at other points within the gear or disc, to oscillate. If the load, speed and other operating variables are held constant, an equilibrium oscillating condition is reached and the surface temperature can be considered to oscillate about some fixed mean. Because any one point on the surface is in the conjunction for an extremely short time compared with the time that it takes to complete one revolution, it is commonly assumed that the surface temperature is nearly equal to the mean over most of the cycle; however, there is a rather sharp rise from the mean to a maximum as the point passes through the conjunction, followed by a rapid decrease owing to conduction and convection.

For rectangular conjunctions, Blok's critical temperature hypothesis may be stated mathematically as follows: $T_c = T_s + \Delta T$, where T_c is the maximum surface temperature in the conjunction, T_s is the mean surface temperature ahead of the conjunction, and ΔT is the maximum temperature rise in the conjunction (sometimes referred to as the flash temperature). According to Blok, scuffing occurs when T_c reaches a critical value, i.e. when $T_{cr} = T_s + \Delta T$ where T_{cr} is the critical temperature. Blok derived an expression for the quantity ΔT and for the general case of two bodies made of different materials (31). When the two bodies are made of the same material, this general expression may be simplified to read (39):

$$T = 0.62\ fW^{3/4}\ (\sqrt{V_1} - \sqrt{V_2})\ R^{-1/4}\ E_r^{\ 1/4}\ b^{-1}$$

where f is the friction coefficient, W is the load per unit width of track, V_1 and V_2 are tangential velocities of the two surfaces relative to the conjunction zone, R is the equivalent radius of curvature of the two bodies of radii R_1 and R_2 forming the conjunction, given by $(R_1^{-1} + R_2^{-1})^{-1}$, E_r is the reduced modulus of elasticity, given by $E(1-v^2)^{-1}$ where E is the modulus of elasticity, v is Poisson's Ratio, and b is Blok's thermal coefficient, given by $(Kpc)^{1/2}$ where K is the thermal conductivity, p the density, and c the specific heat (46).

4.1.2.2 Material Strength Related Failure Modes

The failure modes to be discussed in this category are Pitting, Plastic Flow, Rippling, Ridging, and Breakage.

4.1.2.2.1 Pitting

Pitting is identified as the formation of pits on metal surfaces. Pitting is the consequence of repeated stress cycling (compression to tension and vice-versa) of the contact surfaces beyond the metals endurance limits, leading to surface or sub-surface cracks with eventual detachment of metal fragments resulting in the formation of pits. Initial pitting occurs within a few hundred cycles of commissioning of new gear pairs due to detachment of projected high points considered manufacturing defects. Progressive pitting takes time and occurs due to metal fatigue. Presence of a lubricant film of adequate thickness modulates stress cycling but does not eliminate it. Therefore, pitting is considered a material fatigue strength related failure rather than lubrication related failure.

Although pitting in nearly pure rolling systems, such as, Rolling element bearings, has received a great deal of attention, pitting in sliding/rolling systems, such as in gears, has so far been largely overlooked. This latter oversight may be due to two reasons. First, as mentioned before, scuffing has an overriding influence on maximum gear performance. Second, with gears of low contact ratios, pitting usually occurs near the pitch line, where the Hertz stress is maximum and the motion is nearly pure rolling (4).

4.1.2.2.2 Plastic Flow

Plastic flow is another material strength related failure. Plastic flow can take several forms but always results from loading the gear material in the contact zone above its yield stress. If compressive loads are high or vibration causes high peak loads (especially if the gears are soft) tooth surfaces can become peened or rolled, much the same as the head of a cold chisel or rivet is peened by repeated blows. Although the cause of failure lies with the material or the loads in the system, a more viscous oil can help to cushion the blows and prevent plastic flow (29).

4.1.2.2.3 Rippling

Rippling is also plastic deformation but is caused by surface shearing stresses rather than compressive stresses. It is likely that these stresses can be lowered by proper lubricant formulation to give low coefficients of friction. Generally, rippling does not lead to immediate failure and may even be advantageous, since the ripples may serve as oil reservoirs on the surface. It is an indication of high loads and may be a warning of future failure (29).

4.1.2.2.4 Ridging

Ridging is plastic flow due to high spots on a gear plowing over the mating surface. Ridging sometimes occurs on hardened hypoid gears, where a lubricant having antiweld or antiscoring properties is required. In a sense, ridging is evidence that the lubricant has been successful in preventing welding at pressures exceeding the yield point of the steel. Under these conditions, it is likely that wear takes place at the same time; thus it may be a form of lubricant failure if the lubricant has good antiweld properties (29).

4.1.2.2.5 Breakage

Gear tooth breakage is the fifth category of material related failures listed by AGMA. True tooth breakage cannot be influencd by the lubricant. It is important to be able to distinguish between breakage failures due to tooth fatigue and breakage failures resulting from pitting or other initial causes. Gear teeth are loaded as cantilever beams, the load being applied at various positions along the contacting face. The shape of gear teeth is such that this applied load causes a maximum bending stress in the metal somewhere in the root area of the tooth and almost invariably below the contacting surface. Thus, a tooth broken off at the root failed in bending; there is no known lubricant that strengthens or weakens gear materials. In some cases of bending fatigue failure, a crack, once started in the root, may propagate upward toward the tip of the tooth. In such cases, the crack can usually be traced to its origin by observing the fracture and noting the "beach" marks. These circular or semicircular ripples are concentric about the origin of the crack and are reliable indicators of the start of failure. If a tooth breaks because of pitting, the fracture will have started from one of the pits near the middle of the tooth, resulting in mid-tooth breakage. Breakage due to overload will not leave characteristic beach marks; rather the fracture surface will usually be quite rough and will originate in the root area.

In some cases, it is possible to detect impending failure by periodic inspection before the evidence of the cause is obliterated by subsequent destruction. Some types of failure, such as pitting and scoring, can eventually lead to secondary failure due to destruction of the involute profile. Proper analysis of a failure very often requires painstaking detective work. The important point is to determine the cause of failure by observation of the gears, the lubricant, and the operating conditions and history of the unit (29).

4.1.3 Gear Lubrication

Gear surfaces deform elastically during the momentary meshing action due to the contact loads coming into play. This elastic deformation alters the theoretically nonconformal surfaces to a degree of conformity, momentarily, particularly under high contact loads. In classical hydrodynamic theories where the lubrication is explained to be due to the formation of films of the lubricant purely by virtue of the viscosity of the lubricant and the operating loads and speeds, the elasticity effects of the surfaces are ignored. All situations where a fluid film ideally/theoretically fails to separate the contact surfaces are termed boundary lubrication conditions, where conditions are assumed to be partly metal to metal contact and partly fluid film. After the elastohydrodynamic concept put forth by Dowson, it is now accepted that between the pure hydrodynamic region and the boundary region there is a transition stage of elastohydrodynamic region where the surface deformations play an important role.

4.1.3.1 Elastohydrodynamic (EHD) Lubrication

An adequate treatment of the EHD concepts has been made by P.M.Ku, et al, in Reference (4). This treatment has been cited generously, with appropriate changes in numbers indicating references to literature cited thereof, in the following paragraphs.

When counterformal bodies are loaded against each other, their surfaces experience significant localized elastic deformations. Elastohydrodynamic lubrication deals with the interaction between the hydrodynamic action of the lubricant and the localized elastic deformations of the surfaces. It basically explains why an intact oil film may exist under certain conditions between highly loaded counterformal bodies (4).

4.1.3.1.1 Theoretical Film Thickness Equation

It has been found, both analytically and experimentally (47) (48) that the oil film thickness in an EHD conjunction is not uniform. Accordingly, the oil film thickness of particular interest is the minimum oil film thickness, because if rubbing contact were to occur, it would be apt to occur where the oil film thickness is the least.

The basic equation for the minimum oil film thickness, for a rectangular EHD conjunction of perfectly smooth surfaces, in a steady-state, flooded, and isothermal flow, has been given in dimensionless form by Dowson (48). This equation may be written in conventional engineering units as follows:

$$h_m = 26.5 \frac{\alpha_o^{0.54} (\mu_o V_t)^{0.7} R^{0.43}}{\omega^{0.13} \overset{*}{E}{}^{0.03}} \tag{A}$$

where h_m = minimum oil film thickness, μ(in.)

α_o = pressure viscosity coefficient of oil at conjunction inlet temperature and near atmospheric pressure, (psi^{-1})

μ_o = absolute viscosity of oil at conjunction inlet temperature and near atmospheric pressure, (cp)

V_t = sum velocity, (ips)

R = equivalent radius of curvature at the conjunction, (in.)

ω = unit normal load, (lb/in)

$\overset{*}{E}$ = equivalent Young's modulus, (psi)

The above equation applies strictly to a sliding rolling system with a rectangular conjunction, provided the other assumptions stated above are met. In practical applications, the conjunction shape is generally not rectangular, but approximately elliptic in shape. If the aspect ratio of the ellipse normal to the motion is large (> 5), little error results from the rectangular assumption. However, if the aspect ratio is small, then the problem becomes more complex. In that event, an approximate correction for the side flow effect, noted by Cheng, may be used (49).

Additionally, the assumption of an isothermal flow process may not be approached in practice, due to heating caused by the viscous shear of the oil in the inlet region. This effect can be quite significant at high sum velocities, particularly when the oil viscosity is high. In that event, another approximate correction for the inlet shear thermal effect, also noted by Cheng, may be applied (50).

One other assumption involved in the derivation of Equation (A) is that the conjunction inlet is "flooded" so as to allow a full hydrodynamic pressure buildup in the inlet region. In practice, this is often not the case; and the conjunction inlet is said to be "starved." The effect can be very significant when the starvation is severe. In that event, another approximate correction for the inlet starvation effect, by Castle and Dowson, may be used (51).

When the side flow, inlet shear thermal, and inlet starvation corrections are applied to Equation (A), the minimum oil film thickness for an elliptic EHD conjunction of perfectly smooth surfaces, in steady-state flow, is obtained as:

$$h'm = 26.5 \frac{\alpha_o^{0.54} (\mu_o V_t)^{0.7} R^{0.43} \phi_s \phi_t \phi_x}{\omega^{0.13} \overset{*}{E}^{0.03}} \qquad (B)$$

where $h'm$ = minimum oil film thickness, μ (in.)

ϕ_s = side flow correction factor.

ϕ_t = inlet shear thermal correction factor.

ϕ_x = inlet starvation correction factor.

4.1.3.1.2 Film Thickness Ratio

It should be emphasized that Equation (B) is quite approximate due to the approximations involved in the derivation of Equation (A) itself and particularly in the derivation of the three correction factors, ϕ_s, ϕ_t, and ϕ_x. Additionally, the equation as such applies only to perfectly smooth surfaces and a steady-state flow process.

It is well known that actual engineering surfaces are not perfectly smooth. Surface roughness and surface texture affect the EHD film development in a complex manner. But, there is as yet no viable way to assess their effects (52-54).

An empirical parameter is often used in practice to indicate, in a very approximate way, whether or not the operation is in the EHD regime, or how deeply the operation penetrates into the boundary lubrication regime. This parameter is defined as:

$$\Lambda = \frac{h'm}{\delta_c} \qquad (C)$$

where Λ = film thickness ratio

$h'm$ = minimum oil film thickness, μ(in.), as given by Equation (B).

δ_c = composite surface roughness of a pair of surfaces, μ(in.)AA

4.1.3.1.3 <u>Application to Gears</u>

In applying Equations (B) and (C) to gear design and performance analysis, it is important to examine how well all of the basic assumptions enumerated above are realized in actual gear tooth action.

Referring first to the derivation of Equation (B), the gear tooth action is certainly not steady-state, and its consequences on EHD film development needs to be considered. There are, for example, the questions of dynamic tooth load and squeeze film effect. Dynamic load as such is probably not a serious obstacle to the use of the EHD film thickness equation, because the equation states that the film thickness is quite insensitive to load. Even the squeeze-film effect due to normal approach of the surfaces does not appear to have a significant impact on EHD film thickness (55). However, regardless of these, the flow through the gear mesh necessarily involves cyclic fluid acceleration and deceleration, and their effect on EHD film thickness is by and large not well understood (4).

Actual gear-mesh conjunctions are generally not strictly rectangular, but elliptic in shape. If the aspect ratio of the ellipse normal to the motion is large, as in the case of spur gears with perfect tooth-to-tooth alignment, little error is expected from the assumption of a rectangular conjunction. However, as mentioned earlier, gears are extremely susceptible to misalignment. The presence of gear tooth misalignment will result in a distorted conjunction ellipse so that a simple side flow correction can no longer be applied (4).

The assumption of an isothermal flow process does not hold for gears, which ordinarily operate through a wide range of sliding velocities in the mesh cycle. Although a correction may be applied for this effect, as mentioned above, the result can still be misleading unless the nonuniform temperature distribution across the inlet film is taken into account (56). Reliable assessment of the temperature gradient across the inlet film, particularly considering sliding and the complex participating flow and heat transfer involved, is, to say the least, no easy task (4).

Due to the action of the gear teeth and the conventional manner oil supply, the state of gear tooth lubrication is probably always starved, or far from the flooded assumption. Although the effect of starvation on film thickness behavior is quite well understood by assuming an arbitrary inlet boundary location and shape with a uniform temperature distribution across the film, these assumptions are not realistic for gears (51). Moreover, there is presently no reliable way to relate the extent of starvation (i.e. the inlet boundary location) to lubricant, design, and operating parameters, even under these idealized conditions (4).

Finally, actual gear tooth surfaces are, of course, not perfectly smooth, but exhibit some typical surface roughness and surface texture. Equation (C) is a very approximate way to provide an indication of the operating lubrication regime, and as such it does not account for the effect of surface texture. However, even if the complications due to surface texture were ignored, the use of a film thickness ratio in the design process can be very misleading. After all, if the composite surface roughness involved is less than, or even about the same order of magnitude as the calculated EHD film thickness, EHD flow as envisioned in the theory no longer prevails, and the validity of Equation (B) and thus that of Equation (C), becomes quite questionable (4).

The above remarks are not intended to minimize the important contributions of the EHD theory to the understanding of the lubrication of counterformal surfaces. It is only that, as knowledge on the details of

EHD lubrication expands, complications begin to emerge and further refinements appear necessary. In particular, once the operation leaves the full EHD lubrication regime, a continuous and undisturbed oil film no longer exists between the mating surfaces. When this happens, some degree of surface-to-surface contact cannot be avoided; and the failure processes are no longer physical in character as implied by the EHD theory, but must be influenced by the chemical interaction that takes place. It has been stated previously that of the three major lubrication related gear tooth failure modes, EHD lubrication is not a necessary condition for pitting and not a sufficient condition for scuffing. Therefore, in assessing the effect of lubrication related failure modes on gear performance, the crucial question is not when and how full EHD film ceases to prevail, but rather when and how the boundary film formed by the oil metal atmosphere interaction ceases to inhibit or minimize surface failures (4).

In order to ensure full EHD lubrication, a film thickness ratio of the order of 2 to 3 is believed necessary (31). This condition must be approached, much of the time, in the operation of gears lubricated with straight mineral oils. Otherwise excessive wear, if not scuffing, is likely to occur. However, unless the choice of gear steel is very unfortunate, it is not such a serious concern even when operating with straight mineral oils, and certainly not with oils which provide significant or substantial scuffing and wear protection. Accordingly, a design based on the assumption of full EHD lubrication of gear teeth is not only unnecessary in the general context; but is, in fact, too conservative from the standpoint of size and weight (4).

Of course, if one wishes arbitrarily to take into account the chemical interaction involved, one could employ a film thickness ratio of less than unity in design. However, this is difficult because one does not know what specific value to assign to the film thickness ratio. In any case, even if one sets out to design gears to operate at some high or low specific value of film thickness ratio, one has no real assurance that such a design condition will indeed be achieved in practice. As mentioned above, the available technique for calculating the EHD film thickness is quite approximate, not to mention that the meaning of the film thickness ratio is questionable in the all important regime where failures are likely to occur (4).

4.1.3.1.4 Transition from EHD to Boundary Lubrication

As the operation of a sliding/rolling system leaves the full EHD regime, the operation enters the rather ill defined micro and partial EHD lubrication, mixed lubrication, and classical boundary lubrication regimes - herein collectively called the boundary lubrication regime for the sake of brevity (4). Surface-to-surface contact then begins to take place, and becomes more severe as the operation penetrates deeper into the boundary lubrication regime. Consequently, rubbing wear becomes inevitable, scuffing becomes a possibility, and pitting becomes more severe. The manifestation of rubbing wear and pitting damages is time-dependent, and their rates of damage depend upon the physical and chemical oil metal atmosphere interactions. The occurrence of scuffing is quite precipitous, but is also controlled by boundary lubrication considerations to be discussed later.

4.1.3.2 Boundary Lubrication

There is considerable argument and lively debate on the boundary lubrication concepts. Many definitions and terminologies have been offered and mechanisms put forward to explain boundary lubrication. With EHD concepts deeply embedded in our minds, it is now difficult to view boundary lubrication as the concern of the chemist in as much as hydrodynamic lubrication is viewed as the concern of the engineer.

At the January 1972 symposium held at NASA - Lewis Research Center, A. R. Landsdown of the Swansea Tribology Center, U. K., mentions the following useful definition of boundary lubrication deduced by D. Tabor (58):

> "That type of lubrication which cannot be attributed to the bulk viscous properties of the lubricant (whether the system is operating under hydrodynamic or elastohydrodynamic conditions) but arises from a specific solid lubricant interaction,"

Landsdown questions the phrase "a specific solid lubricant interaction" used by Tabor. According to Landsdown, this phrase implies that a single phenomenon may be dominant, which, though could be true under laboratory conditions and simplified models, does not explain this complex phenomena in practical lubrication systems involving steel on steel or ferrous on nonferrous surfaces, with commercial lubricants in an uncontrolled atmospheric environment. Clarifying further, states:

> "The above definition implies that boundary lubrication is necessarily associated with the presence of a liquid lubricant, and this seems to be common usage, in spite of the fact that there are many common features in boundary lubricated (liquid), dry lubricated, and unlubricated systems. In general, we should perhaps think in broader terms about boundary lubrication to insure that we are not placing any artificial limits on our thinking, but in the context of a symposium on liquid lubricants, the usual interpretation is satisfactory."

In the discussion following Landsdown's introductory paper, R. L. Johnson of NASA - Lewis Research Center has suggested that the more positively stated definition for boundary lubrication in the 1969 OECD glossary of terms and definitions on friction, wear, and lubrication be considered as follows (58):

"A condition of lubrication in which the friction and wear between two surfaces in relative motion are determined by the properties of the surfaces and by properties of the lubricant other than bulk viscosity."

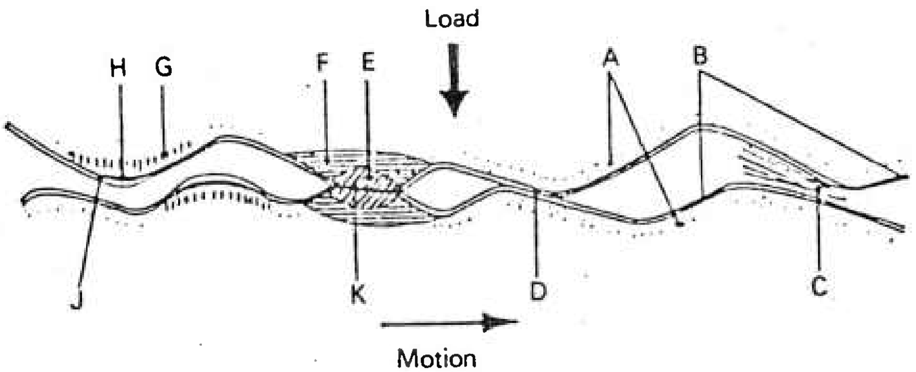
Sir William Hardy (59) adopted the term 'boundary lubrication' from Osborne Reynolds in 1919 to describe the phenomenon at the interface or boundary of a metal and lubricant (60).

For the purpose of this feasibility study, the 1969 OECD glossary definition mentioned by Johnson above is sufficient and covers the scope of investigations. Since a thorough understanding of the various phenomena that can occur when surfaces interact in the presence of a liquid lubricant is important, especially to evolve new criteria for characterising lubricants, a further elucidation of the boundary lubrication concepts as outlined by Landsdown will be adopted here, generously, with appropriate changes in numbers indicating references to literature cited thereof (58).

4.1.3.2.1 The Boundary Stystem

Figure 4-5 shows diagramatically some of the features that may be involved in the boundary system. Even this is a simplified model because it ignores factors such as metal transfer between surfaces, the competition between reagents for both stable and freshly exposed surfaces, and the possible occurrence of special activation phenomena such as exoelectrons (58).

It is a natural consequence of this complexity that the study of boundary systems tends to be fragmentary. For really fundamental study, the systems have to be simplified to the point where the relevance of the results to practical systems is often difficult to establish.



A: enrichment of oxygen and other "contaminants" near metal surfaces;

B: absorbed films of boundary lubricant;

C: viscosity increase adjacent to metal surfaces;

D: absorbed films carrying load between asperities;

E: plastically deformed tip of asperity;

F: elastic deformation of substrate;

G: residual stress in asperity after plastic deformation;

H: work-hardened tip of asperity;

J: chemisorption on abraded surface after contact;

K: local high temperature at asperity contact.

(Reference 58)

Figure 5. The Boundary System

For practical studies, as in the development of new antiwear and extreme pressure (EP) additives, the theoretical basis is weak, and even the relevance of the empirical tests that are used is open to question. In fact, the only justification for most of these studies is that they seem to work (58).

4.1.3.2.1.1 Viscosity Increase Adjacent to Surfaces and Soap Formation

On the different phenomena highlighted in Figure 4-5, the one that has been hotly disputed is item C, the increase in viscosity which has been described in the vicinity of bearing surfaces. The long series of studies of thick lubricant films in oils containing polar molecules, ranging from Hardy (60) to Fuks (61) and Cameron (62) was reviewed in 1969 by Hayward and Isdale (63) who ascribe all the evidence to the presence of particulate contaminants. More recently, Smith and Cameron (64) have produced further evidence for the existence of order in a solvent containing a long chain fatty acid to a distance of 2100Å in an experiment at room temperature. The effect is ascribed to the formation of soaps by reaction between the acid and the steel surface, followed by the entrainment of solvent to produce a pseudogrease structure. If this explanation is correct, then the reaction kinetics of the soap formation and the stability of the agglomerated soap molecules will repay further study. There are obvious objections to the hypothesis that reaction between free fatty acids in solution and the metal surfaces can lead to soap films that are more than monomolecular in thickness. One objection is that under certain sliding conditions, sufficient to disrupt the grease-like structure but insufficient to abrade or scuff the mating surfaces, there should apparently be a disturbance of the equilibrium, resulting in very rapid continuing solution of metal and formation of soap. Such a process should be easy to demonstrate and should in fact have been known already in practical bearing systems. The amounts of soap that have been detected in wear debris (65) does not appear to be as great as would be implied by the ready formation of the multimolecular layers postulated by Cameron (58).

There is some evidence from spectrographic oil analysis programs that part of the wear metals can exist in true solution in the lubricant, even when the lubricant is a plain mineral oil. This requires the formation of some form of organometallic compound, and because of the widespread occurrence of fatty acids, the formation of soaps is an obvious possibility. On the other hand, iron soaps are generally insoluable in common solvents, and some form of micellar dispersion may be a more likely mechanism (58).

4.1.3.2.1.2 Enrichment of Oxygen and Other Contaminants

The presence of oxygen and/or water appears to be necessary for the formation of iron soaps. The contribution of oxygen to the action of other additives is less clear. It has long been thought that most chemical reactions will occur in the presence of very small quantities of such things as oxygen or water, which act as promoters or catalysts. On the other hand, Fein has suggested that atomically clean metals react with almost anything they contact and instanced the case of adhesion of insoluble metals under space conditions (65). It is open to argument whether this adhesion is really analogous to chemical reaction but Morecraft (67) has shown that octadecane, butane, and decoic acid all reacted to give hydrogen, methane, and carbon monoxide when in contact with clean iron surfaces. It is generally accepted that freshly exposed clean metal surfaces are in highly reactive state, but this condition is sometimes believed to be a transient one (58).

4.1.3.2.1.3 Adsorbed and Reacted Films and Their Strength

Commenting further on adsorbed and reacted films, Landsdown says:

> "At the 1967 symposium in San Antonio, Godfrey (68) emphasized the physical aspects of boundary lubricant films and concentrated on the strength and rupture of adsorbed and reacted films. The general assumption was that a

> uniform (usually monomolecular) film forms on the whole of the available surface and that it is the integrity of this film which prevents metal to metal contact, the film being ruptured only when very high normal or tangential stresses arise."

This mechanism may well apply to adsorbed films of the so called mild EP additives such as vegetable oils, fatty acids, and soaps, but the work of Forbes (69) with electron probe microanalysis of wear scars has indicated that this is not the case with the more powerful chemisorbed additives. Forbes infers that where smooth sliding is occuring, the concentration of additive is low, but that there is an increased concentration on torn or otherwise damaged parts of the surface. The implication is that reaction takes place preferentially at freshly exposed surfaces and that the product of reaction insures a nondisruptive operation until the damaged area has been smoothened, when the additive concentration will again be low (58)."

4.1.3.2.1.4 Chemical Activity of Freshly Exposed Surfaces.

There is, of course, a great deal of evidence for such preferential reactions at freshly exposed surfaces. One possible example is that described by Moore in which lithium nitrite acted as an oxidant in greases for use in helium atmosphere (70). Nitrites are traditionally reducing agents, rather than oxidants, and their oxidizing action in this application requires temperatures very much higher than expected or the existence of some alternative activation mechanism (58).

While it is generally accepted that freshly exposed surfaces have a high reactivity, the effect and importance of this in specific wear situations is less easy to predict. Rozeanu (71) demonstrated an abnormal surface potential on fresh fracture surfaces, and it would be interesting if a similar technique could be applied to freshly worn surfaces (58).

4.1.3.2.2 Scope and Change in Emphasis of Boundary Lubrication

Commenting further on the scope and change in emphasis in the interpretation of boundary lubrication concepts, Landsdown says:

> "It is interesting to trace the changes that have taken place in the interpretation of the term boundary lubrication. Originally the distinction was made between hydrodynamic and boundary lubrication, and it was assumed that where hydrodynamic pressures were insufficient to account for satisfactory lubrication, the effect was due to the presence of (probably uniform) adsorbed films. The subject of lubrication was therefore divided, with hydrodynamic lubrication being the concern of the engineer, and boundary lubrication that of the chemist." (58)

4.1.3.2.3 Boundary and EHD Films

With the development of a satisfactory EHD theory, much that was previously considered to be boundary lubrication became also the concern of the engineer. Boundary lubrication was still associated with the existence of uniform adsorbed films (68) but the question was asked (72) whether the physical adsorption of Hardy and Langmuir was important mechanism, or whether the action was chemical, producing reaction products giving a form of rheodynamic or EHD lubrication. Tallian (73) described partial EHD effects in which much of the load is transmitted through an EHD film, but asperities penetrate the film and contact each other. His comments seemed to support the rheodynamic view, but use of the phrase a boundary film implied the assumption of a uniformly sorbed layer (58).

4.1.4 Gear Lubricant Additives

4.1.4.1 Role of Extreme Pressure (EP) Additive Chemicals

Additives have to be used which are capable of working at the high temperatures where mineral oils and mineral oil/fatty oil blends are unsuitable. Phosphorous, chlorine and sulphur compounds are commonly employed and so are sulphurized fat and leads (5).

Reference has already been made to the polar activity of fats and it is probable that the free fatty acid performs two functions, it enables the blend to spread better and in addition, forms a metallic soap. This soap acts as a lubricant and is of low shear strength. Above some temperature, the soap melts or decomposes and is no longer effective. This temperature is not the same with all materials nor is it definitely known for any one material. Temperatures up to 120°C have been mentioned as the limit for metal soaps so formed. The low shear strength of the soap results in low coefficient of friction (5).

Phosphorous additive materials function by combining with the tooth surface material to form phosphides. It has been said that they are only useful in reducing wear when the surface finish is better than 10 micro inches. Where the surface finish is inferior to 10 micro inches, it appears that wear results. In time, of course, the asperities will be reduced in height when the phosphides can then become wear reducers. In other words, controlled wear has been effected. The phosphorous compounds can be active up to about 200°C, depending upon the materials concerned (5).

Chlorinated additives are used extensively. They function by forming low shear strength metallic chlorides which prevent metal contact. They are effective within a range of about 150°C to 400°C. Below 150°C, the chlorides do not form and above the upper limiting temperature they melt or decompose. They reduce the coefficient of friction to a mere fraction of that generated by metal to metal contact (5).

Sulphur is also used in many forms as a gear oil additive and forms metallic sulphides which have a lower shear strength than the original metal. They are not so effective in reducing friction as the salts produced by chlorine and the other materials but nevertheless, the coefficient of friction of the sulphides is considerably less than that of the original metal. The sulphides formed are thought to be most effective between about 200°C to 800°C. Below 160°C, no reaction occurs and above 800°C, melting or decomposition occurs. Again, not all materials behave in the same way. Sulphur and the sulphides formed from it have a functional temperature range depending upon the gear tooth material (5).

Reference has also been made to lead. This is usually employed as a lead soap which is effective up to about 100°C above which it melts or decomposes (5).

The subject of additives is very complex and from the comments made it will be apparent that there is no point in using sulphur, for instance, if only light loads and low temperatures are expected. Indeed an oil containing sulphur only would need to operate in conditions giving temperatures up to 200°C in the contact area before the sulphur could perform its function. In other words, metal-to-metal contact, and hence wear would be necessary to bring sulphur into operation. The sulphide film formed would have a coefficient of friction higher than that of a correctly formulated product for that application (5).

It will be seen too that there is a case for using additives in combination with each other, for instance, phosphorous in conjunction with chlorine and sulphur is suited for temperatures up to 800°C. Only the reactions necessary to accommodate the conditions of operation will take place (5).

While temperatures of operation have been quoted, they are not to be taken as exact, but to illustrate the relative behavior of certain materials used to supplement the natural lubricating ability of mineral oil. It is important that all additives, when in use, do the job for which they are designed at gear tooth contact and do not have adverse effects on other parts of the equipment (5).

4.1.4.2 Film Strength Additives

The E.P. and antiwear additives mentioned in the previous paragraphs generally refer to industrial and automotive gear oils formulated with mineral oils. The E.P. additives that are specific to aviation synthetic lubricants have been discussed and outlined by S. Staley (74) during discussion of the additives paper by S. V. Smalheer (75) at the January 1972 symposium on Interdiciplinary Approach to Liquid Lubricant Technology, NASA - Lewis Research Center, Cleveland, Ohio (R58). This section will be quoted here generously with appropriate changes in reference to literature.

The load-carrying requirements of gas turbine engine lubricants in the past have not been particularly onerous, except for the case of turbo propeller engines where a higher load-carrying capacity was specified to adequately lubricate the reduction gear boxes. As mentioned previously, this aspect was taken care of by the use of a more viscous lubricant (7.5 cs at 210^{o}F compared with 3 cs for pure turbojet lubricants). In the vast majority of lubricants, therefore, additives having only mild E.P. properties have proved quite satisfactory, and tricresyl phosphate in particular has found very wide use. Other phosphate ester variants which have found use include triphenyl phosphorothionate (76) and di-o-chlorphenyl phenyl phosphate (77) while phosphonates, aminophosphonates (78) (79), phosphoramidates, and phosphites are other phosphorous compounds that have found past use. Di-iso-propyl phosphite today finds use as an E.P. additive in diester-based lubricants for other applications; e.g. MIL-L-46000 (74).

Apart from the phosphorus containing compounds that tend to dominate this field, other products that have been used include chlorinated diphenyls, diaryl thioethers (80), amides of hydrogenated dimer acids (81), and even small amounts of dicarboxylic acids (82), such as sebacic acid and azelaic acid (74).

Tricresyl phosphate is still widely used in the modern, so called type 2 lubricants based on esters of trimethylolpropane and pentaerythritol. However, more demanding load-carrying requirements are being presented by engines such as the Olympus 593 of the Concorde (83). The Olympus specification calls for a 5 cs oil with load-carrying properties equivalent to those of 7.5 cs lubricants. Much activity, therefore, currently centers around the development of new and more effective additives. The main difficulty is that the more active load-carrying additives, such as the sulfur and chlorine containing products, normally have a serious adverse effect on oil stability and tend to be corrosive at high temperatures. As with antioxidants, answers have been found in the development of highly synergistic additive combinations. These may, for instance, consist of the relatively stable tricresyl phosphate type together with a small amount of a second, much less stable additive, that activates the tricresyl phosphate. These combinations are proprietary secrets. However, the following examples from the patent literature can be given (74):

(1) Neutral triorgano phosphate plus a neutral salt of dialkyl hydrogen phosphate (84)

(2) Neutral triorgano phosphate and dialkyl hydrogen phosphite (85)

(3) Trihydrocarbyl phosphate plus a salt of an alkylamine, a monohaloalkyl phosphonic acid, and a dicarboxylic acid (86).

It should be noted that in the above treatment of film strength additives for aircraft lubricants, the role of antioxidants, rust, and corrosion inhibitors, which are used to enhance oxidation, and rust inhibition properties to meet qualification requirements have not been discussed as this is beyond the scope of present discussions on enhancement of film strength property.

4.1.5 Lubricant Testing

Lubricant testing can be categorized in four general requirement areas:

- Development/Research
- Qualification
- Quality Assurance
- Troubleshooting

4.1.5.1 Development/Research

Lubricant development/research testing is the most demanding test requirement. Each of the remaining requirements utilize segment/segments of the development sequence. A development sequence starts with fundamental mechanism testing and follows through full field performance testing. This requirement involves the development of a new/improved lubricant for a specified application. It involves considerable time, equipment, and cost investments.

4.1.5.2 Qualification

As the name implies, this test requirement involves the qualification of a lubricant for a prescribed application. This requirement is probably the second most demanding test sequence. It relies on a portion of the test sequence implemented under a development program.

4.1.5.3 Quality Assurance

For each product, the manufacturers and users generally establish a routine lubricant test requirement to ensure that every batch of manufactured product meets a set of quality assurance criteria. These tests can only assure that a new lubricant batch is as good/better than a previous batch, but does not ensure the performance of the product.

This test sequence relies on a portion of the qualification test sequence.

4.1.5.4 Troubleshooting

The last major lubricant test requirement is troubleshooting. These are tests designed to troubleshoot problems especially during introduction of new formulations. Periodic field oil evaluation/examination tests and tests to establish relubrication or oil change criteria fall into this category.

4.1.5.5 Test Classification

Lubricant tests can be classified into two categories:

- Physicochemical Testing
- Performance Tests

4.1.5.5.1 Physicochemical Tests

Physiocochemical lubricant testing includes such tests as viscosity, flash point, specific gravity, color, composition, contamination, pour point, compatability, penetration, and drop point. These tests are utilized for lubricant identification, qualification, troubleshooting, and quality assurance purposes. Such tests are relatively inexpensive, straightforward, and their application is well defined.

4.1.5.5.2 Performance Tests

Performance lubrication tests include such tests as thermal stability, oxidation, corrosion, film strength, and wear. These tests are utilized for lubricant development, qualification, quality assurance, and troubleshooting purposes. These tests are relatively expensive.

There currently exists an extensive proliferation of performance test techniques, approaches, and sequences. These numerous test alternatives vary considerably with respect to time required to perform them, cost of materials/spares, hardware, and required skill levels.

4.1.5.6 Test Levels

Performance tests for lubricants can be grouped into several levels of testing:

- Mechanism Testing
- Component Testing
- System Testing
- Field Testing

4.1.5.6.1 Mechanism Testing

Mechanism tests are basic film strength apparatus tests where the performance of the lubricant is studied in basic geometrical configurations and simple mechanisms. Generally, the performance in such mechanism tests are of a qualitative nature and relate to the configuration of the mechanism. Examples are Four Ball test, Timken test, Falex test, Almen test, etc. While some of these tests have good repeatability/reproducibility, they are not as critical as component and system tests in stating the performance of a lubricant. They are, however, important for initial screening during lubricant development/research and as quality assurance tools.

4.1.5.6.2 Component Testing

Component tests are those where the test specimen is a complete mechanism like a gear or ball bearing, unlike simple geometric forms used in mechanism tests. By designing component tests, metallurgical simulation and application conformity can be achieved. The trend is more and more in favor of such tests to come as close as possible to the application requirements. Examples are Ryder test, FZG test, Rolling Bearing Performance test, etc.

4.1.5.6.3 System Testing

System tests are those where the performance of the lubricant is asessed simultaneously in the utilization of various components involved in a total system, representing the typical application of the lubricant. Examples are pump tests for hydraulic oils, engine tests for crankcase oils, etc. In typical engine test, the crank case oil will be assessed for its ability to perform in the piston ring/cylinder liner zone, in the crankshaft bearings and in the cam and lifter locations simultaneously. System tests are closer to applications than mechanism and component tests.

4.1.5.6.4 Field Testing

Finally, field tests are tests performed using full scale field equipment under a set of controlled and carefully monitored operating conditions/cycles typical of the real life situation to which the lubricant will be subjected in practice. For example, for railroad engine oils, marine cylinder oils, etc., such tests have become mandatory before equipment manufacturers approval can be secured.

4.1.5.7 Test Sequence

Mechanism tests are less time consuming and costly to perform than component tests, which in turn are less time consuming/costly than system tests. Field tests are generally tailor made for a set of equipment/product and are not required where well defined system tests take care of application requirements. The general sequence followed is: the simple, less expensive tests are utilized for screening of candidate formulations and the component/system tests for specification/qualification of selected packages/formulations.

4.1.5.8 Test Characteristics

In each test program and test procedure, several parameters determine the efficiency/expediency with which a lubricant is characterized/rated in reference to an alternate product, a reference product or pass-fail criteria. These parameters include:

- Repeatability
- Reproducibility
- Time
- Skill Level
- Criticality
- Confidence Level
- Field Correlation
- Availability of Hardware
- Equipment/Hardware Cost
- Test Cycle/Life Cycle Cost
- Cost of Spares/Negligence Standards/Standard Specimens.

Repeatability of a performance test refers to the range or spread which one test result established with one equipment by a single operator in a given laboratory can be duplicated a second time under the same operator/equipment and laboratory conditions. This is indicated in each test procedure developed by statistical methods. The repeatability with which a test can characterize, differentiate between and rate lubricants has a bearing on how efficient is the test procedure/result.

Similarly, reproducibility refers to the range or spread within which a test result can be duplicated when identical tests are performed by a different laboratory/equipment operator. Generally, when comparing the performance efficiency of a lubricant with another brand of the same, the repeatability/reproducibility of the test procedure used for characterization is kept in mind.

The time required to complete a performance test presently can range from a few minutes to several days depending on the level of testing (defined later) involved. Test design programs become expensive when testing time is long and the skill level of technicians/operators needed to perform them increases.

The criticality of a test depends on how well the test characterizes a lubricant based on its application requirements. Many times it depends on how far the lubricant users/manufacturers have adopted it in their specifications. For example, a shear stability test for lubricating oil may be more critical than viscosity index for high pressure-low temperature application. But, if industry has not established the procedures/hardware for evaluating shear stability, the test may become less critical than viscosity index. Although it is difficult to establish order or level of criticality for all performance tests, some are more critical than others.

The confidence level at which meaningful comparisons can be made between comparable data decides the discriminating ability of a test. Generally, 95% confidence level is prescribed in most performance tests and the spread or scatter of results at this level of confidence is accepted and taken note of.

Where there are too many unidentified parameters influencing the results, the confidence levels goes down. For the most common types of distributions encountered, 95% of the results fall within one standard deviation of the mean ($\pm$S), and 95% of the results fall within two standard deviations of the mean ($\pm$2S). When prescribing pass-fail criteria, the possible spread of results in a given test procedure due to the inherent limitations (repeatability and reproducibility) of the test should be taken into account.

Field correlation involves the ability to extrapolate test results into expected field performance. This correlation is poor with respect to most test approaches.

Factors like availability of hardware to perform the tests, equipment/hardware costs, test cycle/life cycle costs, cost of spares or reference standards, and cost of test specimens also deterime the efficiency and expediency with which a test can be performed.

4.2 Ryder Gear Test

The Ryder Gear Test is a performance test described in ANSI/ASTM D-1947, standard method for load-carrying capacity of petroleum oil and synthetic fluid gear lubricants (1). The oil (or fluid) under test is evaluated in a standard gear machine, at a series of increasing loads, under semi-controlled conditions. The amount of tooth face scuffing occurring at each load increment is measured. The percentage of tooth face scuffing is plotted against the respective load to determine the load-carrying capacity of the test oil (or fluid). Load-carrying capacity of the test oil (or fluid) is the tooth load, in pounds per inch of tooth face width, at which an average tooth face scuffing of 22.5% has been reached.

4.2.1 Test Heads

Three test rigs have been approved for use under the test. Table 4-1 gives the general description of the Ryder, the WADD, and the EAF (AFB) test heads, any of which could be used for the ASTM D-1947 test.

The Erdco Universal tester is comprised of a drive system, a support oil system, a test oil system, and necessary instrumentation and controls (1). This tester is described as universal in that the drive system may be used to drive several lubricant test devices. The Ryder gear machine operates on the power circulating principle; two parallel shafts are connected by two slave gears and two test gears to form a square so that the power required to operate the machine is that required to overcome the friction losses in the gears and bearings (four square principle). The operating principle of the WADD gear machine is identical to that of the Ryder gear machine. However, improvements in material and design permit its operation at speeds up to 30,000 r.p.m., and test gear temperatures of 700°F (370°C) or higher (1). The WADD gear machine differs from the Ryder gear machine in that the two shafts are supported by two double row roller bearings instead of three journal bearings. Screw thread type nonrubbing seals, rather than elastomer seals, are used to separate the test oil and support oil chambers (1).

It should be noted from Table 4-1 that there are significant variations between two of the three test heads in the mounting of test gears and seal types and between all the three test heads in drive shaft and driven shaft assemblies. The machine constants also differ significantly. In short, each of these three test heads should be viewed as a separate machine, while analyzing precision data in view of the influence of deflections and alignment discussed in Sections 4.1.1.4 and 4.1.1.5.

The Reference Oil C Report, as discussed in Section 2.1, lists in detail the load-carrying capacity data gathered from 1965-1970 with each of these three test heads (3). This data, however, does not list the reject tests where repeatability stipulations of the ASTM D-1947 test have not been complied with. Table 4-2 gives an overall summary of Ryder data for Reference Oil C obtained in the above program at the

TABLE 4-1 ***

GENERAL DESCRIPTION OF TEST HEADS

	Ryder	WADD	EAF **
Case Material	SAE 122 Cast Iron	SAE H-11 Modified Tool Steel*	SAE 122 Cast Iron
Shaft Material	AMS 6260	AMS 6260	AMS 6260
Slave Gear Helix Angle, Degrees	13.7291	13.7291	13.7291
Bearings, Number and type:			
Drive Shaft	3 Journal	2 Double-Row Roller and 1 Ball Thrust	1 Single-Row Roller and 2 Ball Thrust
Driven shaft	3 Journal	2 Double-Row Roller	2 Single-Row Roller
Test Gears Mounted On Overhung Shafts	No	Yes	Yes
Seals, Number and Type Between Support and Test Section	4 Elastomer Lip Seals	2 Nonrubbing Screw-Thread Air Seals	2 Nonrubbing Screw-Thread Air Seals
Seal, Load Chamber Type	Piston Ring	Nonrubbing Labyrinth	Nonrubbing Labyrinth
Load Piston Area, Sq. In.	4.54	2.76	2.43
Machine Constant	18.55	11.50	9.96

* Latrobe Steel VDC hot work tool steel.
** EAF test head is also abbreviated as AFB test head.
*** From Table III - 1 of reference 2.

TABLE 4-2 *

SUMMARY OF REFERENCE OIL C MEAN
LOAD-CARRYING CAPACITY RESULTS OBTAINED
USING RYDER GEAR MACHINES

(1966, 1967, 1968, 1969, 1970)

Laboratory	No. of Determinations						Mean Load Carrying Capacity, lb/in.					
	1966	1967	1968	1969	1970	Total	1966	1967	1968	1969	1970	Over-all
SwRI	54	24	28	38	28	172	2710	2670	2875	2800	2913	2802
A	6	-	4	-	-	10	3119	-	3540	-	-	3287
B	44	28	25	20	6	123	2909	2829	3018	2828	3022	2905
C	-	8	10	22	25	65	-	3010	2686	2708	2879	2808
D	4	-	-	-	-	4	2724	-	-	-	-	2724
F	30	17	18	32	36	133	2791	3003	3067	2908	3125	2974
G	54	56	64	134	80	388	3051	3069	3242	2972	2947	3036
Seven Labs	192	133	149	246	175	895	2877	2934	3085	2914	2972	2949

Laboratory	Repeatability Standard Deviation						Repeatability Standard Deviation ÷ Mean, %					
	1966	1967	1968	1969	1970	Over-all	1966	1967	1968	1969	1970	Over-all
SwRI	207	228	266	261	258	257	7.6	8.5	9.2	9.1	8.9	9.2
A	141	-	296	-	- -	296	4.5	-	3.4	-	-	9.0
B	289	318	192	230	161	272	9.9	11.2	6.4	8.1	5.3	9.4
C	-	273	288	240	241	271	-	9.1	10.7	8.9	8.4	9.6
D	10	-	-	-	-	10	0.4	-	-	-	-	0.4
F	342	199	285	322	230	309	12.2	6.6	9.3	11.1	7.4	10.4
G	209	232	381	278	160	282	6.8	7.6	11.8	9.4	5.4	9.3
Seven Labs	250	251	317	275	207	279	8.7	8.6	10.3	9.4	7.0	9.5

Reproducibility Standard Deviation = 337 lb/in
Reproducibility Standard Deviation ÷ Overall Mean = 11.4%
95 Percent Confidence Interval of Overall Mean (Based Upon the Reproducibility Standard Deviation) = ± 22 lb/in

* From Table 8 of Reference 3

participating seven laboratories. The variation of mean load-carrying capacity has been beyond the prescribed limits of 2760 to 3160 lb./in. in many cases. Also, the overall repeatability standard deviations have been higher, in some cases, than the 95% confidence limit of 284 lb./in. indicated in the test procedure. These deviations are much more pronounced if compared with similar data on WADD and EAF (AFB) test machines, which are summarized in Table 4-3.

The data compiled under the CRC program, as discussed in Section 2.2, is summarized in Table 4-4 (2). It should be remembered that data is from one laboratory only and, although the average mean load-carrying capacity of Reference Oil C has been within the prescribed limits, the standard deviation figures within and between the three machines employed are significant for one laboratory.

It is therefore concluded that the precision problem is not just a reproducibility problem between laboratories, as concluded in the Reference Oil C Report, but also a repeatability problem between different test heads from data gathered under the CRC program (2) (3).

4.2.2 Variables Affecting the Ryder Gear Ratings

In the CRC program, an attempt has been made to identify some of the variables affecting load-carrying capacity ratings obtained in the ASTM D-1947 test (2).

The variables investigated in the CRC program are:

- Reference Fluid (Reference Oil C and Hercolube A)
- Test Heads (Ryder, WADD, and EAF)
- Surface Finish of Test Gears
- Tip Relief of Test Gears
- Test Gear Spline Internal Diameter
- Test Gear Hardness
- Metal Monitor Readings of Test Gears

TABLE 4-3 *

SUMMARY OF REFERENCE OIL C MEAN LOAD-CARRYING CAPACITY RESULTS REPORTED USING WADD AND AFB GEAR MACHINES **

(1966, 1967, 1968, 1969, 1970)

Laboratory	No. Of Determinations						Mean Load Carrying Capacity, lb/in.					
	1966	1967	1968	1969	1970	Total	1966	1967	1968	1969	1970	Over-all
SwRI[a]	24	8	14	18	4	68	2778	2908	3109	2860	2800	2884
C[b]	37	16	-	8	14	75	2936	2861	-	3021	3180	2975
D[b]	2	4	-	-	-	6	2581	3256	-	-	-	3031
E[a]	19	12	22	12	4	69	3182	3277	3255	3002	3229	3193
Four Labs	82	40	36	38	22	218	2938	3035	3198	2939	3120	3017

Laboratory	Repeatability Standard Deviation						Repeatability Standard Deviation ÷ Mean, %					
	1966	1967	1968	1969	1970	Over-all	1966	1967	1968	1969	1970	Over-all
SwRI[a]	434	196	277	221	122	334	15.6	6.7	8.9	7.7	4.4	11.5
C[b]	247	221	-	381	391	303	8.4	7.7	-	12.6	12.3	10.2
D[b]	0	156	-	-	-	369	0.0	4.8	-	-	-	12.2
E[a]	302	201	331	314	109	300	9.5	6.1	10.2	10.4	3.4	9.4
Four Labs	324	206	312	290	330	314	11.0	6.8	9.8	9.9	10.6	10.4

Reproducibility Standard Deviation = 340 lb/in.
Reproducibility Standard Deviation ÷ Mean = 11.3%
95 Percent Confidence Interval of Overall Mean (Based Upon the Reproducibility Standard Deviation) = ± 45 lb/in.
(a) Data Obtained Using WADD Gear Machine
(b) Data Obtained Using AFB Gear Machine

* From Table 9 of Reference 3
** The AFB test head is also abbreviated as EAF test head.

TABLE 4-4 *

TEST RESULTS SUMMARY
(CRC PROGRAM)

TEST VARIABLE	AVERAGE LB/IN	STANDARD DEVIATION LB/IN	NUMBER OF OBSERVATIONS
Test Head Type			
Ryder	2583	483	32
WADD	2571	654	32
EAF	2493	744	32
Lubricant			
Ref Oil C	3046	470	48
Synthetic	2051	279	48
Gear Mfg. Period			
Early	2366	498	48
Later	2732	698	48

* From Table V-3 of reference 2.

4.2.2.1 Reference Fluid

Two types of reference fluids (one of mineral oil base/MIL-L-6082C and the other of synthetic base/Hercolube A) were used in the program. Properties of these fluids are outlined in Appendix F.

4.2.2.2 Test Heads

The three different test heads (Ryder, WADD, and EAF) described under Section 4.2.1 were used at the Alcor Inc. Laboratory where the tests were conducted.

4.2.2.3 Surface Finish of Test Gears

Taking the narrow and wide test gears together and grouping surface finish of both sides of all test gears used, the average surface finish of gears varied from 22.0 to 38.2 in.CLA.

4.2.2.4 Tip Relief of Test Gears

The tip relief variations for the test gears used ranged from 0.0001 inch to 0.0006 inch on the average.

4.2.2.5 Test Gear Spline Internal Diameter

The variation in this parameter was from 1.3568 inch to 1.3580 inch between both sides of the test gears.

4.2.2.6 Test Gear Hardness

Taking narrow and wide gears together, the average Rockwell C hardness of the test gears varied from 61 to 66.

4.2.2.7 Metal Monitor Readings of Test Gears

Taking narrow and wide gears together, the average metal monitor readings varied from 0.5 to 5.4.

4.2.3 Correlation of Ryder Gear Test Variables With Load-Carrying Capacity

Correlation coefficients obtained in the CRC program betwen varibles listed in 4.2.2 and the average load-carrying capacity (L.C.C.) have been listed in Table 4-5. The correlation coefficients indicated against each variable has been worked out by GE by statistical techniques using data generated in the CRC program. The closer the absolute value of these coefficients to unity, the more linear is their influence on load-carrying capacity. If the coefficient indicated is negative (-) it means that an increase in the absolute value of that parameter leads to a decrease in the load-carrying capacity rating and if the coefficient is positive (+), it means that an increase in the absolute value of that parameter leads to an increase in the load-carrying capacity rating.

A quick look at Table 4-5 shows that the change over from one test head to another has very little influence on load-carrying capacity when any one reference oil is rated.

Of the five test gear factors; namely, surface finish, tip relief, spline I.D., hardness, and metal monitor readings considered, surface finish of the wide gear, tip relief of the narrow gear, and hardness of both gears influence load-carrying capacity much more than some of the other parameters indicated. None of these parameters however, have been widely varied in the experiments conducted due to the obvious limitation of availability of test gears with widely varying dimensional and metallurgical properties. Also, most of the correlation factors are indicating nonlinear correlations (far from unity).

TABLE 4-5 *

CORRELATION COEFFICIENTS

Load Carrying Capacity Correlation With Respect To:	Equation Factor Coefficient
Lubricant (Ref Oil C vs Hercolube)	0.793
Test Head	
(WADD vs Ryder)	0.025
(EAF vs Ryder)	-0.063
Lubricant/Test Head Interaction	
(WADD vs Ryder)	0.416
(EAF vs Ryder)	0.396
Surface Finish	
Narrow Gear	-0.176
Wide Gear	-0.264
Tip Relief	
Narrow Gear	0.241
Wide Gear	0.168
Spline I.D.	
Narrow Gear	-0.130
Wide Gear	-0.226
Hardness	
Narrow Gear	-0.252
Wide Gear	-0.292
Metal Monitor**	
Narrow Gear	-0.025
Wide Gear	0.146

* From Table V-7 of Reference 2

** Metal monitor is a gross measurement of metallurgy by electronic means. A constant value suggests uniform metallurgy; but it can also result from compensating changes in metallurgy.

Reviewing the work of Kelly (38) and Carper (87), reported findings in this area are disturbing. According to Kelly, an increase of surface roughness of 10μin. RMS, decreases load-carrying capacity by nearly 25%. Carper has also reported from previous unpublished SWRI data on the Ryder test, that a tip relief of only 0.0004 in., which is within the manufacturing tolerance of the Ryder test gears, increases the load-carrying capacity by nearly 50%.

One other important test gear/configuration variable which is very important is misalignment of the test gears. In this study, specific experimental data could not be gathered on the same. However, it has been reported (87) that Kelly's findings (38) have shown that a misalignment of only 0.005 radians decreases the load-carrying capacity by almost 35%.

4.2.3.1 Tip Relief and Misalignment

The above findings are important in the sense, no matter how much one controls operation test parameters like load, speed, etc., some of the latent and uncontrollable parameters like tip relief and misalignment could drastically affect load-carrying capacity in gear test rigs. This has been one of the important reasons for the present search for alternative techniques other than gear rigs.

4.2.4 General Industry Feelings Concerning the Ryder Gear Test

In additon to analyzing specific Ryder Gear Test data treated under the previous sections, discussions were held with laboratories and individuals connected with this test, both in industry and government. Some of the comments/observations that have been gathered are summarized below:

- Lack of correlation between laboratory results and field application.

• The geometry, surface finish, spacing and type (spur) of gears are not truly representative of present gears utilized in the field.

• Response to changes in lubricant composition (additives) formulation requires considerable testing work and thus time and money.

• Lack of confidence in test results has discouraged use of Ryder Gear Test among the major lubricant suppliers and thus has seriously impacted future research efforts.

• Increasing cost of test specimens in relation to the value and utilization of the data obtained.

• Firm requirement for a high precision, low cost lubricant load-carrying capacity determination technique.

• Lack of enthusiam among new developers of synthetic lubricants in securing qualification approvals due to the increased costs of testing.

The above comments/observations are self explanatory and need no further comment.

4.2.5 Direction of Improvements to the Ryder Gear Test

A listing of directional Ryder improvement approaches indicated by analysis are presented as follows:

• Improved gear failure characterization criteria from the present 22.5% scuffing load criteria due to its heavy dependence on operator judgement (especially while estimating area scuffed).

• Increased temperature control of test specimens and bulk oil.

• Increased test gear stress control (load and speed).

• Real time and more precise assessment of wear.

• A redesign of the test specimen configuration to a more realistic state.

• Tighter control of manufacturing tolerances of test gears, particularly in respect to tip relief and surface finish.

On improved characterization criteria, data has been gathered from the findings of the SWRI study (4). Table 4-6 gives the 22.5% scuff limit as well as 10% scuff limit failure loads obtained for a range of test speeds.

In order to examine the range of spread in the results at each test speed and characterization criterion, standard deviation figures were derived for each set of 8 determinations indicated in Table 4-6. The data at 5000, 10,000, and 15,000 rpm indicate a reduction in standard deviation figures (i.e. less scatter) as the characterization criterion is changed from the 22.5% scuffing load to 10% scuffing load. However, this possible trend is disrupted at the 2,500 r.p.m. where the effect is reverse. If one particular rating of 4,220 lb./in., which is too far away from the mean value of 3684 lb./in., is considered removed and deleted, the standard deviation value for the remaining 7 results at 2,500 r.p.m. falls in line with the possible trend indicated (as criterion is changed from 22.5% scuffing load to the 10% scuffing load). Unfortunately, the initial scoring load which occurs prior to even the 10% scuffing load has not been determined in these experiments. As will be shown later the initial scoring load is more indicative of film failure than percent area related scuffing load. Also, the initial scoring load is less influenced by operator judgement which is generously exercised in estimating area scuffed. In the present Ryder test, however, there is no provision for determining initial scoring load. Thus, the SWRI results indicate that the characterization criteria employed to determine failure load influences precision.

TABLE 4-6 **

GEAR SCUFFING TEST RESULTS (RYDER)

(1973-1976)

Gear Speed (rpm)	Scuff Limited Load-Carrying Capacity, lb./in. At 10% Average Scuff			At 22.5% Average Scuff		
2,500	3,550 3,210 3,680 4,220*	3,650 3,755 3,815 3,595	Standard Deviation: 269* Mean: 3,684	3,970 4,080 4,000 4,540	3,990 3,950 4,120 4,270	Standard Deviation: 188 Mean: 4,115
5,000	2,700 2,750 2,980 3,185	3,205 2,930 3,250 3,275	Standard Deviation: 213 Mean: 3,034	3,000 2,940 3,520 3,440	3,800 3,320 3,570 3,540	Standard Deviation: 274 Mean: 3,391
10,000	2,570 2,370 2,470 2,410	2,405 2,535 2,395 2,130	Standard Deviation: 125 Mean: 2,411	2,760 2,570 2,690 2,720	2,590 2,850 2,510 2,400	Standard Deviation: 136 Mean: 2,636
15,000	1,715 1,820 1,785 1,830	2,130 1,985 2,110 1,995	Standard Deviation: 143 Mean: 1,921	2,040 2,060 1,990 2,090	2,420 2,230 2,360 2,310	Standard Deviation: 153 Mean: 2,188

* The high standard deviation figure of 269 with 10% average scuff data at 2,500 has been due to the value 4,220 obtained in comparison with average value of 3684.

If 4,220 is considered a freak result and left out, the standard deviation for the other 7 results would be 182 at the 2,500 rpm gear speed for 10% average scuff criterion.

** Data from Table 10 of Reference 4

Increased temperature control of test specimens during a test is a difficult proposition as the point of action (rubbing) where the maximum heat is generated is far away from the core in a test gear. Internal cooling of gear surfaces is difficult to design and achieve.

Manufacturers of Ryder Test Gears (P&W aircraft) will probably not respond favorably to any impositions of better manufacturing tolerances especially with respect to tip relief and surface finish at the tooth and flank areas. Even if they can accomplish this, it would increase costs per specimen which will further increase the cost of testing, already considered expensive by oil formulators and suppliers.

4.3 Alternative Gear Rigs and Other Film Strength Test Apparatus

In the field of lubricant technology, performance evaluation techniques to assess the capabilities of a lubricant in reducing frictional resistance, wear and surface distress of interacting surfaces in relative motion has grown into a specialized area. Most assessments relating to the ability of a lubricant to provide a film strong and thick enough to separate the load bearing surfaces in sliding/rolling motions are termed film strength tests and the apparatus for such tests come in a variety of forms. Although in a broad sense, the physico chemical attributes of a lubricant are expected to indicate its ability to perform, its real accomplishments in generating a film to carry the load under dynamic conditions depends on a number of system and operating factors like: geometry of contact, metallurgy and surface texture of interaction, operating parameters such as loads, speeds and temperature as well as the environment surrounding the interfaces. Theoretical hydrodynamic, boundary, and EHD concepts as described in Section 3.0 have been postulated to predict the influence of and interdependence of these operating and system parameters responsible for generation and stability of lubricant films. For example, in contraformal geometries like meshing gears, cams, ball and roller bearings, etc., the viscosity of a lubricant is not as helpful as in conformal geometries like plain bearings for securing a film capable of separating the contacting asperities and supporting the applied load. Similarily, operating loads and speeds can generate films in certain regimes but can destroy them when these regimes are not encountered or exceeded. The surface temperatures of contacting geometries can be helpful in releasing the chemistry of lubricants when maintained at certain levels but can decompose films formed when those temperatures are exceeded. Metallurgies like chromium and nickel are generally nonreactive to sulphur-phosphorus chemistry of lubricants, while other metallurgies like copper, silver, and brass could be over reactive and strong EP chemistry could prove to be a disadvantage in the latter case. Therefore, film strength property of a lubricant cannot be predicted by one or more of the physico chemical properties like viscosity or level of additive chemistry in lubricants. Lubricant performance is as perceived and not inherent and as such there are no simple "litmus" tests to performance characterization. At the same time, the choice of a proper film strength test for performance characterization is important, since what is perceived in a certain combination of motion kinetics, and operating parameters may not be perceived in actual applications unless the film strength test truly simulates the rolling/sliding motion, the metallurgy, the contact stresses and the temperature conditions prevelant in the actual application. Film strength performance or load-carrying capacity of a lubricant therefore depends on the way it is characterized, the method of assessment, and the operating and system parameters of the test design.

Some of the film strength apparatus developed and being developed for lubricant evaluations other than the Ryder, WADD, and ERDCO machines are listed as follows:

A. GEAR RIGS

NASA Lewis Gear Fatigue Tester
IAE Gear Machine (IP-166)
FZG Gear Machine (DIN 51 345)
Four Square Gear Machine (Gleason Machine)

B. DISC MACHINES

Edge to Edge Rig
AFAPL Disc Tester
Caterpillar Disc Machine
SAE Machine (Ring-Machine)
Amsler Disc Machine
Mobil Coryton Disc Machine/Rig

C. OTHER POINT AND LINE CONTACT MACHINES

Timken Machine/Rig (ASTM D-2782, 2509)
Falex Machine/Rig (ASTM D-3233)
Four Ball EP & Wear Tester (ASTM D-2783, 2596)
Almen Machine (Reference 8)

In the first phase of this feasibility study, an attempt was made to gather as much information as possible on some of the above film strength apparatus and other gear rigs. The information collected regarding their system and operational features have been reported in matrix form in Appendices A to C. In many cases, complete information was not available. On machines listed under (C) above, information is available in the ASTM or other references indicated in parenthesis and as such, these were not presented in matrix form. In Appendices A and B, important operational test specimen and hardware information concerning gear machines (A) listed above have been indicated in matrix form. In Appendix C similar data has been presented for some of the disc machines (B) listed above.

A comparison of the relative merits of some of these alternate machines/film strength testers and their suitability as alternate testers to the Ryder gear machine will be discussed under section 5.0.

5.0 PHASE II, ANALYSIS OF SURVEY

The terminology developed under Section 4.1 on Boundary, EHD, and gear lubrication concepts, discussion of the Ryder precision problem, and information on alternative film strength tests form the background for the analysis of the data base carried out in this section.

The objective of Phase II is to perform an in-depth analysis of potential Ryder Gear Test modifications and existing alternate test hardware. Potential new characterization concepts resulting from such analysis will also be discussed.

5.1 Ryder Gear Test Modification

5.1.1 Ryder Hardware

The results of the CRC program and the related work of Kelly and Carper, as previously described, have focused attention on three important variables which must be controlled in a gear test such as the Ryder. These are stated in their order of importance as follows:

- Tip Relief of Test Gears. A 0.0004 in. variation can affect L.C.C. ratings by as much as 50%.

- Misalignment of Test Gears. A variation of 0.005 radians can affect L.C.C. ratings by as much as 35%.

- Surface Finish of Test Gears. A 10 μin. RMS variation can affect L.C.C. by as much as 25%.

In Section 4.2.3 it was pointed out that tip relief of gears used in the CRC program was susceptible to such variations in excess of 0.0004 in. between individual test gears.

5.1.2 Rating Criteria

Another major element concerning Ryder precision is the scuffing/scoring criteria presently utilized. As discussed in the previous section, the SWRI Study (4) has indicated a significant reduction in test result data standard deviation as the characterization criteria was changed from 22.5% scuffing load to 10% scuffing load. The present criteria approach also is very labor intensive.

5.1.3 Modification Feasibility

Based on the substantial hardware and criteria modifications required for Ryder precision improvement, such a modification does not appear cost effective.

5.2 Existing Alternative Test Techniques

Much has been written about the relative merits of the various tribo-testing machines and whether the information obtained from their use is worthwhile (29). With the exception of gear and disc machines listed under Section 4.3, none of the other film strength machines simulate the type of motion (sliding and rolling with low slide/roll ratio) encountered in meshing gears. Any alternative test technique selected to replace the Ryder test should not overlook the primary purpose of having a gear rig test as a qualification test, particularly in aviation synthetic oil specifications. The main and auxiliary power transmission gear boxes of military as well as commercial aircraft generally operate at high speeds and high load/unit weight due to weight reduction considerations. In fact, a basic consideration in the design of aircraft engine accessory and main power transmission gear boxes is lubricant load-carrying ability rating (88). The type of sliding and rolling motion at low slide/roll ratios, encountered in Ryder gears for example when operating at 10,000 r.p.m., cannot be provided by machines other than gear or disc. Secondly, nondisc or gear machines do not necessarily rate lubricants in the same order (29). Table 5-7 gives the rating obtained

TABLE 5-7*

COMPARATIVE RATING OF LUBRICANTS ON FILM STRENGTH MACHINES

Lubricant	Four-Ball		Timken		SAE
	Seizure Load	Final Pressure	OK Load	OK Pressure	Failure
Synthetic + Phosphorus	4	1	1	1	1
Mineral + Sulfur	5	5	2	1	3
Mineral + S—Cl	1	2-	3	3	4
Mineral + S—Pb Soap	2	4	4	7	2
Special Lube	2	2	5	5	5
Olive Oil	7	6	6	3	7
Mineral Oil	6	7	6	6	6

* Data from Table 8 of Reference 29 (page 27-6).

on lubricants of different performance levels and Table 5-8 gives an indication of the poor correlation of machines like the four-ball with gear machines like the IAE and hypoid. Therefore, unless the alternate test selected simulates meshing action of gears, one will not have the confidence to use their data to qualify oils for gear lubrication.

5.2.1 Comparison of Film Strength Machines With Gear Rigs

Correlation of film strength machines with gear machines is a critical consideration. Much work has undoubtedly been carried out in the field, but there are only a few references in the literature. Reference (90) concludes that there is little correlation between film strength testers and four square gear machine results (29).

Of the several prominent gear test machines listed in section 4.3, the IAE and FZG machines have been compared on operational parameters in Table 5-9, with the Ryder machine. Table 5-10 gives some comparative test results with different additive packages employed for automotive gear oils. This table shows the poor correlation of Four-Ball, Falex, and Timken machines with IAE and FZG machines.

Although, in this study, specific data on correlations between film strength machines with the Ryder Gear Test have not been gathered, it can be speculated that such correlations, (order of ratings) would be unlikely, in view of lack of correlations between four square machines like IAE/FZG gear rigs with Falex, Four Ball, and Timken machines as indicated in Table 5-10.

5.2.2 Comparison of Gear Rigs With Disc Machines

Generally speaking, disc machines are considered to simulate the sliding/rolling motions encountered in gears and a carefully designed disc machine such as the Mobil Coryton machine has rated lubricants in the same order as an IAE gear machine (91). Table 5-11 gives the details of the lubricants evaluated and Table 5-12 gives the order of rating in

TABLE 5-8*

CORRELATION OF FOUR-BALL MACHINE WITH GEAR RIGS

Four-Ball Criterion	Degree of Correlation With	
	IAE, %	Hypoid, %
Initial Seizure Load--10 in	26	32
Initial Seizure Load--60 in	21	28
$2^1/_2$-in. SD	14	27
Flash-Temperature Parameter	24	22
Mean Hertz Load	35	24

* Data from Table 9 of Reference 29 (page 27-7).

TABLE 5-9*

GEAR RIG TESTS: COMPARATIVE OPERATING DATA

Rig Data	IAE Machine	Ryder Machine	FZG Machine
Pinion speed 'n' (rev/min)	2000, 4000, 6000	10,000	2175, 4350
Test Oil Flow (pt/min)	$^1/_2$, 1, 1	0.476	Dip Lubrication
Test Oil Temp (^oC)	60, 70, 110	74	90
No. of Tests	4	4	1
Initial Load	10 lb	370 lb/in^2 (5 lb/in^2 gauge)	22 lb
Loading Sequence	5 lb Increments	5 lb/in^2 Increments	Uniform Increments of 20 kg/mm^2 Herzian Stress
	5 min run	10 min run	15 min run ('A' Gear) 7.5 min run ('C' Gear)
	5 min rest	10 min rest	Unspecified rest
Failure Load	Lever load (W) at which at least 60 percent of both face/flank combinations scuffed or scored.	The pitch line loading (R) at which 22.5 percent total tooth area is scuffed.	The normal load (W_N) at which the change to high wear range occurs.
Pitch Line Load (lb/in) (tangential)	61 W	R	1.169 W_N

* From Table 2 of Reference 30.

TABLE 5-10**

CORRELATIONS: COMPARISON OF GEAR MACHINES WITH FILM STRENGTH MACHINES

Additive	IAE Safety Load lb	FZG 12th Stage*	Four-Ball Mean Hertz Load, kg	Falex Load lb	Timken Load lb
Base oil	37.5	Fail	33.6	500	5
Lead Soap/ Sulphur-Chlorine	95	Pass	78.5	2500	65
Lead Soap/ Active Sulphur (MIL-L-2105)	-	Pass	100.6	4500	25
Sulphur-Phosphorus (MIL-L-2105B)	80	Pass	68	1750	45
Sulphur-Chlorine-phosphorus	105	Pass	72	4500	60

* Maximum

** From Table 4 of Reference 30.

TABLE 5-11*

TEST LUBRICANTS FOR IAE GEAR MACHINE/DISC MACHINE COMPARISON**

Lubricant	Viscosity (cs) 100 F	210 F
A. Straight Mineral Oil, Paraffinic	102	10.6
B. G. M. Dexron Automatic Transmission Fluid	41	7.2
C. Ford Type Automatic Transmission Fluid (M 2C 33 E/F)	40.8	7.8
D. SAE 20 W30 Tractor Oil Universal (Engine, Transmission, Final Drive Hydraulics)	74	10
E. Multipurpose Transmission/ Wet Brake Tractor OIl	100	10.5
F. Automotive Gear Oil Meeting MIL-L-2105 Specification	91.9	10

* From Table 2 of Reference 91.

** The disc machine referred to is the Mobil Coryton Rig (Appendix C)

TABLE 5-12*

COMPARISON OF IAE GEAR MACHINE AND DISC MACHINE TEST RESULTS

Oil	Failure Loads (lb/in)			
	Disc Machine**		IAE Gear Machine	
	50 C	80 C	50C	110 C
A	3,200	<1,600	4,280	1,870
B	1,600	<1,600	5,350	3,740
C	4,000	<1,600	6,950	4,820
D	3,600	2,800	5,620	7,220
E	3,200	<1,600	5,080	5,600
F	9,200	6,400	8,000	<10,700

* From Table 3 of Reference 91.

** The disc machine referred to is the Mobil Coryton Rig (Appendix C) the disc machine and IAE machine in this study (91). Unfortunately, data comparing results of Ryder Gear rig with those of disc machines on the same formulations/test oil, could not be gathered for this study.

From test data gathered on various alternative test rigs, it is concluded that disc machines have the best chance of rating lubricants in the same order as gear machines with the remaining types of film strength rigs being poor in such correlations.

5.3 New Characterization Concept

The limited studies made on characterization of scuffing load in the Ryder Gear Test (see Section 4.2.4) and a survey of alternate characterization criteria used in various film strength testers, Table 5-13, indicates considerable room for development of new criteria to charcterize load-carrying capacity in the boundary lubrication regime. In the following sections, the approach towards a new concept development will be described.

5.3.1 Criteria for Characterizing Load-Carrying Capacity

The boundary lubrication regime is governed by mixed regimes of partial metal-to-metal contact and partial thin film lubrication where asperities are separated by a lubricant film. Asperities come in contact over their peaks and get separated over their valleys. The applied load is therefore carried by both the area represented by the contacting asperities and the area represented by the thin film regions. As long as there is no softening and plastic flow at the contacting asperities, the performance is smooth and the only distress to surfaces is reflected in the adhesive wear of the asperities which come into contact. The frictional resistance is a sum of the resistance to motion at the contacting asperities and the resistance to motion, due to the viscosity of the film at these asperities separated by the lubricant. As the applied load is gradually increased, a stage is reached where the thin films also rupture and there is a sudden increase in the intensity of stresses

TABLE 5-13

ANALYSIS OF LUBRICANT PERFORMANCE CHARACTERIZATION
BY PRESENT TECHNIQUES/CRITERIA

Test Hardware (Test Technique)	Ref	Lubricant Performance Property Assessed	Characterization Criteria			
			Fatigue	Wear	Seizure Load	Post Seizure Load
1. FOUR BALL TESTER (ASTM D 2783, 2596)	1	a. Load Wear Index b. Last non-seizure load c. Weld Point		X	 X	 X
2. TIMKEN TESTER (ASTM D 2782, 2509)	1	a. OK Value b. Score Value			X X	
3. ERDCO/RYDER/WADD (ASTM D 1947)	1	Load Carrying Capacity				X
4. IAE GEAR M/C (IP-166)	IP Standards	a. Initial Scuffing Load b. Load Carrying Capacity			X	 X
5. FZG GEAR M/C (DIN 51 354)	DIN Standards	Pass Load (stage)		X	X	
6. AFAPL (USA)	87, 89	Scuffing Load			X	
7. AMSLER MACHINE (Switzerland)	93	a. Anti Wear Property b. Scuffing Load		X	 X	
8. MOBIL CORYTON RIG (U.K.)	91	Scuffing Load			X	
9. ROLLER TESTER (Germany)	94	Pitting Load	X			
10. CATERPILLAR DISC TESTER (USA)	87, 95	Scuffing Load			X	

at the contacting asperities which are called upon to bear the entire load. The result is sudden softening, seizure and rupture of the same, bringing about a surface distress recognized as scoring. Continued increase of applied load or continuation of the operations under the scoring load brings about further rapid destruction of the contacting asperities and this is recognized as scuffing which is nothing but an advanced form of scoring.

In the present Ryder Gear Test (ASTM D-1947), the test procedure consists of running the standard gear specimens for intervals of 10 minutes, at each of the predetermined applied loads, to see if the scoring/scuffing damage manifests itself by a process of accumulation or build-up of a distress related function capable of rupturing the lubricant film supporting the load. In other words, at each of the 10 minute observation cycles, it is assumed that there is a distress related function which grows with time and that a 10 minute interval is sufficient to see if this function reaches a critical value at which scoring/scuffing occurs.

A study of gear failure modes (see Section 4.1.2) such as wear, scoring, scuffing, pitting, and tooth breakage, identifiable in lubricated gear contacts, indicates that only pre-scoring wear and progressive pitting are truly time dependent phenomena. Each of these modes of distress increases proportionately with time and the rate of such increase depends on the applied load. Although scoring and scuffing are considered lubrication related, they occur quite suddenly without any noticeable indications prior to the surface damage indicative of scoring. None of the theoretical models such as Archard's, Blok's, and Dowson's postulated to date to explain Boundary and EHD lubrication phenomena, account for a time element that explains build-up of distress modes like scoring and scuffing. Blok's hypothesis, which is considered to offer the best rationale to explain scoring and its accumulated form scuffing, accounts for a load dependent build-up of temperature, which upon reaching a critical value characteristic of the lubricant, material, and operating conditions, ruptures the lubricant film partly supporting the applied load, resulting in softening, welding, and tearing of the contacting asperities which bear the entire applied load. The EHD concept accounts for the existence of a lubricant film of minimum thickness greater than the combined height of the contacting asperities which when ruptured or reduced result in scoring and scuffing. This film is load, speed, lubricant, and material related. Existence of a time element is not indicated in this model.

The first evidence of scoring of contacting test specimens in film strength test apparatus, as indicated by sudden appearance of rough spots due to localized softening, welding, and tearing of contacting asperities, is in itself an indication of the failure of the lubricant film to support/carry contact load. This, in turn, stresses the contacting asperities beyond their plastic deformation point leading to localized softening, welding, and tearing, characteristic of scoring damage. At loads exceeding the scoring load, the contacting surfaces will be operating under post seizure conditions of wear and tear where rapid and uncontrollable changes will be taking place in the surface texture (such as roughness) and material properties (such as hardness and thermal conductivity) of the standards (specimens) themselves. At a load necessary to give $22^1/_2$% scuffed area, it is postulated that the so called load-carrying capacity of the lubricant has been far exceeded and is more indicative of the material resistance property under post seizure conditions of stressing of the specimens than a property indicative of the capacity of the lubricant to function.

It is therefore concluded, that the present criteria for characterizing the so called load-carrying capacity (L.C.C.) of a lubricant as applied in the Ryder Test (ASTM D-1947) is not only arbitrary as stated in the test itself, but is also questionable. This ambiguity in the charaterization process is one of the prime reasons for lack of precision in scuffing load determinations (other prime parameters affecting precision are operational and system parameters relating to the test and have been discussed previously). There is, therefore, a need for a better characterization procedure and a quest for a distress mode that accumulates and aggrevates with continuous change in load and time period, through which the load is applied and contact surfaces are stressed.

After onset of scoring, if a lubricant is able to act swiftly to heal the softened asperities and restore smooth operations and control accumulation and growth of further scoring damage, then that action is indicative of the lubricants ability to function in the post scoring regime of operations, at the applied or operating load at which a distress suddenly manifests.

A further review (see Table 5-13) of various criteria used in ASTM/IP/DIN standards in test rigs like the Four-Ball, Timken/IAE/FZG, and similar hardware indicate that several other criteria like wear rate, initial scoring load, rise in friction, post scoring recovery time, have all been used with some success to characterize the lubricants capacity to carry load and prevent or at least control surface distress.

Only in gear rigs like the Ryder and IAE are the results characterized by the post scoring load in which damage to a prescribed area of apparent contact is taken as the distress indicative of failure of the load-carrying capacity of the lubricant, thus stressing the standard specimens to loads exceeding the scoring load.

5.4 Parameters Influencing Precision in Film Strength Testing

In Section 4.2.1, several system variables affecting precision measurement of load-carrying capacity were discussed. In this section, the general parameters affecting precision in film failure characterizations will be considered. This will be applicable to most film strength tests where load-carrying capacity is assessed on wear, initial seizure load, or post seizure load criteria.

In the Dowson-Higginson equation for calculating the film thickness under EHD lubrication, several parameters responsible for formation of the film were identified. Since failure of the film formed is considered as the prime reason for lack of load-carrying capacity and development of a distress mode such as scoring in our analysis of criteria (see section 5.3.1) these very parameters can be examined for their relative influence on film thickness ratio "Λ". This gives a good indication of whether the surfaces are ideally separated or operating under conditions favoring scoring of the contacting surfaces. Some of these parameters have been listed in Table 5-14. Their relative influence can be examined by analyzing the numerical value of the exponents to which each parameter is raised in the EHD equation. Thus, if "Λ" = 1, is taken as the ideal condition for a scoring damage to occur, then h'm and δ_c have a direct influence on the accuracy or precision with which scoring can be characterized. But, h'm (minimum film thickness) is related to several operational and system parameters such as μ_0, α_0, etc. listed in Table 5-14. Thus, the latent (hidden) operational coefficients ϕ_s, ϕ_x, ϕ_t have a major influence in deciding the value of h'm since their exponents are equal to 1, but $\overset{*}{E}$ (Equivalent Youngs Modulus) and W_N (unit normal load) have relatively smaller influence in film thickness estimations since their exponents are 0.03 and 0.13 respectively in the EHD equation. Similarly, we may say that V_t (sum velocity of sliding) has a greater influence on film thickness than R (equivalent radius of curvature) because of the respective numerical values of their exponents.

5.4.1 Relative Influence of Precision Parameters

In Table 5-15, the above concept has been developed into a relative influence scale of 0 to 100 where 100 indicates direct influence and 0 indicates no influence. Where the influence is not clearly indicated or shown in the three different theoretical models examined in Table 5-14, the rating given is left blank. Also, these influences have been separately examined for the three different characterization criteria, namely

TABLE 5-14

ANALYSIS OF THEORETICAL MODELS TO UNDERSTAND PARAMETERS AFFECTING PRECISION IN FILM STRENGTH TESTING

MODEL/ CONCEPTS	FORMULA	SYMBOLS	COMMENTS
Dowson's EHD Concept	$h'm = \frac{26.5\alpha_o^{0.54}(\mu_o Vt)^{0.7}R^{.43}\phi_s\phi_t\phi_x}{W_n^{0.13}\ {*E}^{0.03}}$ $\Lambda = \frac{h'm}{\delta c}$	h'm = minimum film thickness μ_o = abs. viscosity of oil α_o = press. viscosity coeff. Vt = sum velocity of sliding R = Equivalent Radius W_n = Unit normal load *E = Equivalent young's modulus Λ = film thickness ratio δc = composite surface roughness ϕ_s = side flow factor ϕ_x = inlet starvation factor ϕ_t = inlet shear thermal factor	Indicates influence of material and operating variables on minimum film required for EHD lubrication. A control of these variables controls friction and wear and prevents seizure at Λ greater than 1.
Blok's Critical Temp. Concept	Tc = Tf + Ts $Tf = \frac{0.6222{*}EfWn^{3/4}(\sqrt{V1} - \sqrt{V2})}{\beta R^{1/4}}$ $e = \sqrt{pck}$ $pf \sim (Vt)^{3.66}$	Ts = quasi-steady surface temp. Tc = critical temp. Tf = flash temp. Tc = quasi-steady surface temp. *E = Equivalent youngs modulus Wn = unit normal load V1,V2 = surface velocities β = Block's thermal coeff. R = equivalent radius of curvature p = density c = sp.ht. k = thermal conductivity Pf = failure load	Indicates influence of material and operating variables affecting Tf and Tc, and Ts which are expected to characterize scuffing phenomena.
Archard's Wear Concept	$V = \frac{KWL}{3H}$	V = wear volume W = normal load L = wear distance H = hardness K = Archard's wear coefficient	Indicates influence of material and operating variables affecting wear of contacting asperities.

TABLE 5-15

ANALYSIS OF RELATIVE INFLUENCE OF PERCEIVED PARAMETERS AFFECTING PRECISION IN PERFORMANCE CHARACTERIZATION OF LUBRICANTS

Perceived Parameters (*)	Symbol	Unit	Relative Influence (**) on Precision Pre-Score Wear	Initial Load	Post-Score Load
1. SYSTEM (***)					
Absolute viscosity of oil	μo	Cp	70	70	?
Pressure viscosity of oil	αo	psi^{-1}	54	54	?
Composite surface roughness of specimens after break-in	σa	μin.	100	100	100
Hardness of specimens (initial)	H	psi	100	100	?
Equivalent radius of specimens in contraformal contacts	R	in.	43	43	25
Equivalent youngs modulus of elasticity of specimens	*E	psi	3	3	25
Density of specimens	p	lb/in^3	?	?	50
Specific heat of specimens	c	in/F	?	?	50
Thermal conductivity of specimen	K	lb.f/Fsec	?	?	50
Coefficient of friction	f		?	?	100
2. OPERATIONAL (***)					
Test duration (Time)	t	mt.	100	100	?
Wear distance	L	in.	100	?	?
Normal load	W	lb.	100	100	75
Unit normal load	W_n	ppi	13	13	75
Unit tangential load	W_t	ppi	?	?	100
Power transmitted	P	hp	?	?	100
Sum velocity of sliding	Vt	ips	70	70	100
Quasi steady surface temp. of specimens	Ts	°F	?	?	100
Lubricant jet temperatures	θ	°F	?	?	100

Note: * Perceived parameters considered are those which have been investigated at one time or other in literature.

** Relative influence has been estimated taking EHD, Blok's and Archard's theoretical models on a scale of 0 to 100. Direct influence = 100; no influence = 0; influence not known = ?.

*** System parameters are those which are likely to change during the process of testing and are influenced by geometrical configuration and metallurgical properties of test specimens. Operational parameters are those relating to running of a given test procedure on which generally good control can be exercised.

wear, initial seizure load, and post seizure load as indicated in Table 5-15. The relative influence scale for post seizure load criteria such as the scuffing load, has been indicated by examining Blok's critical temperature concept and the equations developed for the critical temperature which offers the best rationale for such failure to occur. A simple wear model such as the one proposed by Archard is considered for examining the wear volume or weight loss due to the rubbing of contacting asperities in boundary lubrication (92).

5.4.2 Summary of Relative Influence Scale

Table 5-15 and the relative influence scale will be useful in understanding the most critical parameters influencing characterization criteria. Thus, for example, in scoring load measurements, hardness and composite roughness are important system parameters to be controlled or specified. Similarly, almost every operational parameter listed in Table 5-15 influences directly the post score load like the scuffing load concept used in Ryder Gear Test and similar tests.

While Tables 5-14 and 5-15 are developed on known theoretical equations, Table 5-16 lists some of the latent (hidden) parameters not effectively accounted for in these equations. For example, misalignment and tip relief factors are not accounted for in the theoretical models considered, but they are known to affect precision measurement o load-carrying capacity in gear test rigs such as the Ryder.

TABLE 5-16

ANALYSIS OF LATENT PARAMETERS (*) INFLUENCING PRECISION IN PERFORMANCE EVALUATION BY MESHING GEAR CONFIGURATIONS

LATENT PARAMETERS	SYMBOL	ASSUMPTIONS MADE IN THEORETICAL ESTIMATES	DEVIATIONS IN GEAR MESH CONFIGURATIONS
Lubricant flow and inlet starvation factor	ϕx	Fully flooded inlet	Never fully flooded and probably always starved
Squeeze film effect and side flow correction factor	ϕs	Ideal continuous flow assumed	Never ideal in practice
Isothermal flow process temp distribution and inlet shear factor	ϕt	Uniform temp distribution across film is assumed	Does not hold good as heating of film is caused by viscous shear at the inlet region
Gear mesh conjunction	B	Rectangular conjunction is assumed	Never strictly rectangular but generally distorted and approximately eliptical due to susceptibility to misalignment/ vibrations
Composite surface roughness after break-in	σa	Assumed perfectly uniform	Never uniform. Use of film thickness ratio in design can be misleading. An increase in roughness by 10 μ in. RMS is reported to increase load carrying capacity by 25%.
Misalignment of Gears	Sm	Perfect alignment assumed	Misalignment of 0.005 radians reportedly decreased load carrying capacity by 35%.
Tip relief	-	--	A tip relief of only 0.0004 in., which is within manf. tolerance of ryder gears, increases load carrying capacity by 50%.

Note: * Latent parameters considered are those which have been mentioned in literature but on which no or little control can be exercised.

6.0 PHASE III - SELECTION

The objective of the third phase is to determine the most promising test alternative based on the comparative analysis developed under Phase II. Alternatives include Ryder modification, existing techniques, and new concepts. A summary of the selection decision process is presented in Figure 6-1.

Based on inputs reviewed under this program, the option of eliminating the load-carrying capacity test is not a viable option. Many organizations have deemphasized the Ryder test as a result of its precision problems and not as an indication of the significance of load-carrying capacity rating.

The second section of this report addresses in detail the precision problem exhibited by the Ryder Test. The extent of this problem coupled with the resulting economic consequences eliminates leaving the Ryder "as is" as a viable program option.

Ryder modification presents itself as the next element in the program decision process. As covered under Section 5.0, required Ryder hardware and criteria modifications would not be cost effective, thus eliminating this as a viable alternative.

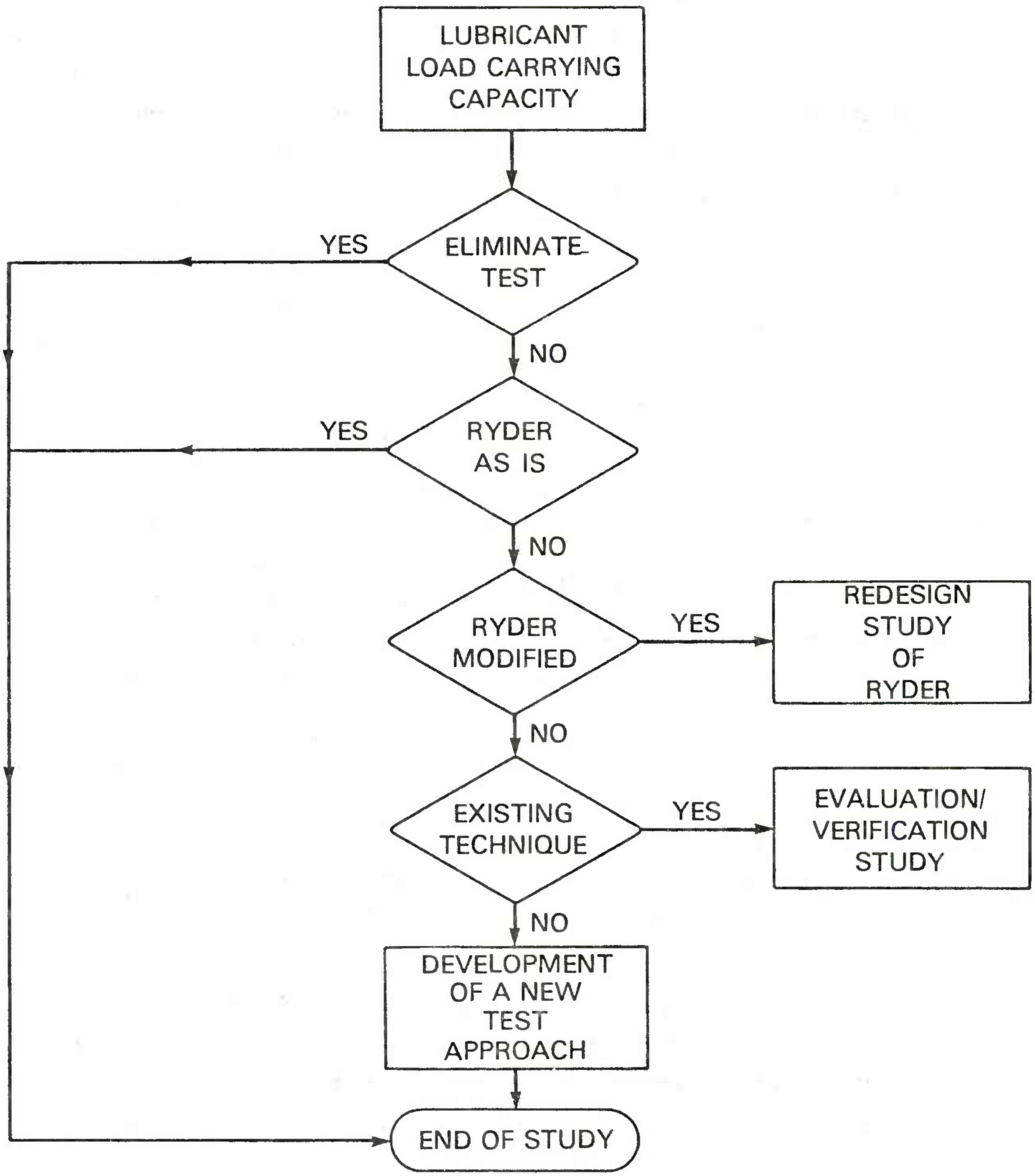
As a result of this elimination process, the two remaining alternatives are utilization of an existing alternative test rig or the development of a new test approach. These remaining alternatives will be addressed in the following sections.

6.1 Test Criteria

Section 5.3.1 addresses how the present Ryder Gear Test characterization criteria is bringing into focus certain post scoring conditions like changes in bulk hardness of the contacting surfaces, changes in surfce roughness, and thermal conductivity. This situation results from continuation of the characterization process into the post seizure

FIGURE 6-1

PROGRAM DECISION PROCESS



regime in order to determine the load at which 22.5% of the apparent contact area is damaged by the scuffing mode of surface distress. It was deduced that scoring and scuffing are not time dependent criteria, and that unless real time monitoring of surfaces is established, precise characterization will be difficult. Under Section 5.3.1, it was presented that the first evidence of scoring of the surfaces is, in itself, indicative of failure of the load-carrying capacity of the lubricant. Also, it was deduced under section 5.4.2 that surface roughness, hardness, and thermal conductivity changes of test specimens operating under the post scoring loads will directly influence characterization criteria. The following new characterization approach is proposed in the light of these findings.

6.1.1 New Criteria

In order to more adequately characterize lubricant performance and to eliminate the influence of arbitrary characterization criteria on the precision of the test, a modified three step approach is proposed for characterizing a lubricant's ability and capacity to perform as follows:

Wear Criterion - To be developed by real time quantitative wear rate assessment by monitoring wear debris for a gradual but continuous enhancement of load intensity with time, till reaching the scoring load.

Scoring Load Criterion - To be developed on a real time basis by monitoring the contact surfaces as well as friction traces for a gradual but continuous enhancement of load with time, till there is a sudden increase in friction with visual evidence and confirmation of score marks on the surface which are monitored on a real time basis.

Post Scoring Recovery Time Criterion - To be developed by continuing the testing process under the scoring load (unlike steps 1 and 2) and determining recovery time, if any, required to arrest continuous rise in frictional resistance in order to achieve steady state conditions in the post seizure regime.

This step would provide additional information about the performance capabilities of lubricant formulations which release their chemistry to restore steady operations though at an increased wear rate, chemical activity, and perhaps noise/vibration levels. For lubricants which do not have such recovery time, there would be continued growth of scoring distress, even at the scoring load at which there was first evidence of softening of the contacting asperities.

This new criteria is represented in Figure 6-2. In lubricant specifications, like the MIL-L-7808, limits are proposed to be introduced on the basis of wear rate, scoring load, and recovery time by examining a number of candidate formulations presently considered satisfactory for aircraft gear lubrication.

6.1.2 Advantages in New Criteria

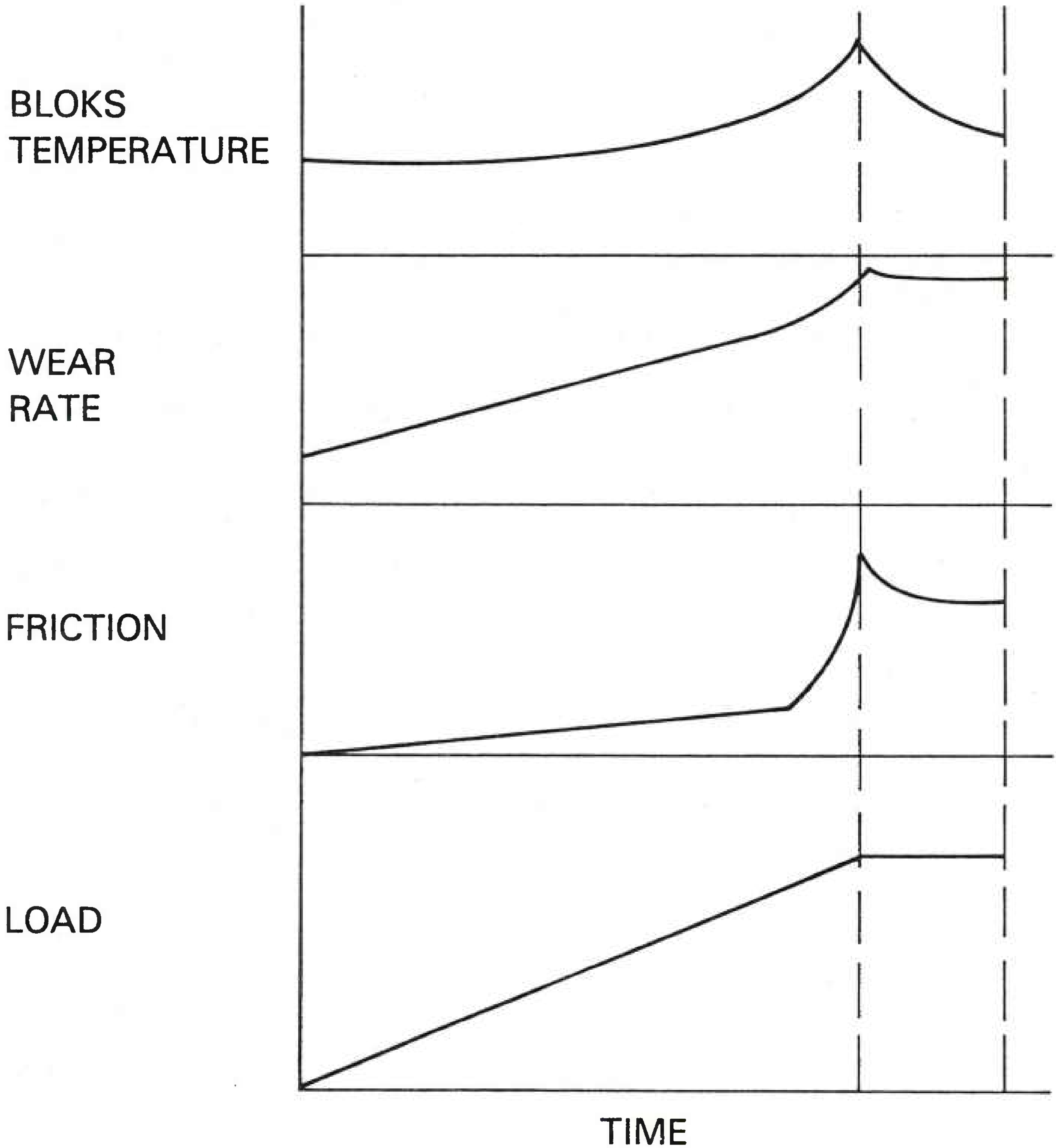
The proposed steps 1 and 2 of new criteria are time and load dependent criteria where load is varied continuously with time, thus eliminating the present ambiguity which is considered partly responsible for lack of precision in the Ryder Test. The new criteria will also be most suited for real time assessment procedures considered necessary to secure better precision and eliminate errors involved in stop-and-observe type of testing where any accumulation of distress related function vanishes the moment the test is stopped and is not carried over for the next load increment. It also eliminates the test duration ambiguity. Step 3, where load is held constant to observe recovery time from scoring, would provide additional information on a real time basis on lubricants which have such a capacity. The recovery time criterion is also a time dependent criterion by which added information about the lubricant is obtained at the distress load.

6.2 Test Configuration

The data developed on alternative existing test devices revealed that basically four types of test configurations are used in film strength tests:

FIGURE 6-2

GRAPHICAL REPRESENTATION OF NEW CHARACTERIZATION CRITERIA



- Point contact configurations like the ball on ball configuration used in a four ball machine, where one of the specimens is stationary.

- Line contact configurations like the disc (cup) on block configuration used in the Timken machine, where one of the specimens is stationary.

- Line contact configurations like the disc on disc configuration where both specimens rotate.

- Line contact configurations like the gear on gear configuration where both specimens rotate.

In order to examine the suitability of such configurations, weighing factors were developed to identify critical examination factors.

6.2.1 Weighing Factors

Alternative test configurations are evaluated with respect to the following factors:

- Gear rigs have precision problems similar to that of the Ryder, due to lack of control on some of the perceived parameters like alignment of the test gears, specifying the roughness of gear profiles, control of tip relief in gear manufacture, and difficulties in controlling specimen temperature during a test, all of which significantly influence precision in load-carrying capacity determinations.

- Any technique selected for gear lubricant evaluation should simulate the type of sliding and rolling motion to which gear profiles are subjected to during contact, and should reflect the low slide/roll ratios encountered in spur, helical, and bevel gears, thus reducing chances of lack of correlation due to improper simulation.

• It should simulate loads/stresses, sliding/rolling speeds, and contact metallurgy encountered in aircraft gear contacts and also the present Ryder Test, in order to cover the performance levels established by the Ryder Test and accustomed to by the present users of aircraft lubricants.

• It should be amenable to real time monitoring techniques, considered necessary for better precision.

• It should give a better hold on other parameters affecting precision, like surface roughness, temperature of specimens during test, and characterization of distress in real time.

• It should be less time consuming than the Ryder Test and reduce testing cost per sample analyzed as compared to the Ryder Test, since lack of precision is resulting in many reject tests for which lubricant formulators, suppliers, and users have to pay.

6.2.2 Selection Process

Table 6-17 outlines the critical parameters on which the decision tables have been based. These critical parameters were selected from considerations of the weighing factors listed in Section 6.2.1.

In this table, test configurations like the four-ball and Timken have been compared with the test configurations of disc and gear machines. Most other configurations like ball on cylinder, cone on ball, pin-on-disc, etc. have motion kinetics similar to the four ball and Timken concepts, since one of the specimens is always fixed with respect to the other in such configurations in a way similar to the four-ball and Timken configurations. Therefore, it is enough if these two ASTM Standard configurations are compared with disc and gear configurations.

Table 6-17 indicates that four ball and Timken configurations cannot give the required type of motion, namely, sliding and rolling, with low

TABLE 6-17

TEST CONFIGURATION DECISION TABLES

A CRITICAL PARAMETERS	B REF	C Desired goal for simulation with AIRCRAFT-TYPE GEARS	D Simulation with RYDER-TEST	E 4-Ball test configuration (ASTM D-2783)	F Cup-on-Block (Timken) Configuration (ASTM D-2782)	G Disc on Disc configuration (CONCEPTUAL)	H Gear on Gear configuration (CONCEPTUAL)
1. TYPE OF MOTION	5	Sliding & rolling as in any spur, helical or bevel gear, operating under low slide/roll ratios	Sliding & rolling as in any spur, helical or bevel gear, operating under low slide/roll ratios	Pure sliding motion for the surfaces of lower 3-balls which are fixed and characterized for surface distress.	Pure sliding motion for the surface of block which is fixed and characterized for surface distress	Sliding & rolling motion for both discs possible Desired low slide/roll ratio can be secured by independently driving the discs and using phasing gears to secure desired surface speeds.	Sliding & rolling motion for both gears possible. Desired low slide/roll ratio can be secured by independently driving the gears and using phasing gears to secure desired surface speeds.
2. VERSATILITY a. Rotation (rpm)	1		10,000 rpm.	Current machines run at 1200/1440/1760 rpm.	Current machines run at 800 rpm.	Desired rotational speeds can be secured to simulate sliding/rolling speeds of Ryder.	Desired rotational speeds can be secured to simulate sliding/rolling speeds of Ryder.
b. Rolling Velocity (ips.) pitch line (Vp)	1		1,833.3 ips	With 1/2" dia. steel balls and for possible 1200 to 1800 rpm., surface speed range is 31 to 46 ips.	With 1.938" dia. cups and for 800 rpm. surface speed possible is 12.8 ips.	Desired low slide/roll ratio is possible.	Desired low slide/roll ratio is possible.
c. Rolling Velocity (across line of action) (ips)	96						
High (V_1)	Ap-D		963.39 ips	-	-	Desired rolling velocity can be simulated	Desired rolling velocity can be simulated.
Low (V_2)	Ap-D		439.79 ips	-	-		

A	B	C	D	E	F	G	H
d. Sliding Velocity (Max)(V_s)	Ap-D		523.60 ips	31 to 46 ips	12.8 ips	Desired sliding speed and low slide/roll ratio is possible to secure by independently driving the discs at desired speeds using phasing gears.	Desired sliding speed and low slide/roll ratio is possible to secure by independently driving the gears at desired speeds using phasing gears.
e. Slide/Roll Ratio	Ap-D		0.5435 ips	Infinite for fixed balls.	Infinite for fixed block.		
f. Hertz Stress (psi)	88,1 Tab.18	125,000-250,000		3,916 to 750,000 for load range 15 to 300 lbs.	63,694-175,438 for load range 5 to 100 lbs.	Desired Hertz stress can be secured by proper choice of load & disc sizes.	Desired Hertz stress can be secured by proper choice of load & gear sizes.
g. Metallurgy	88,1	AMS 6260, 6265	AMS 6260	Steel balls are generally EN-31/SAE 52100	Presently carburized steel but desired metallurgy can be secured.	AMS 6260/6265 can be secured.	AMS 6260/6265 can be secured.
h. Composite roughness	4		Desired 9 to 14.5 μin.AA.	Cannot be specified due to difficulty in measurement.	Can be specified for Block and Cup.	Can be specified for the disc pair.	Can be specified for the gear pair but difficult to measure.
i. Recycling of Test Specimens			Not Possible	Not Possible	Possible	Possible	Not Possible
CONTROL (On parameters affecting precision)							
a. Surface Roughness of Specimens (after break-in) (μ-in)	87		Currently no limits imposed.	Difficult to control as measurement is difficult.	Cup and block can be finished to desired roughness using standard grinding wheels/silicone carbide paper and then run-in by a standard procedure.	Discs can be finished to desired roughness using standard grinding wheels/silicone carbide paper and then run-in by a standard procedure.	Due to the involute profile, finishing, and run-in to desired roughness is difficult.
b. Initial Hardness of Surfaces	4		Desirable to have Case 81-84 (R_a) Core 35-40 (R_c)	Case 64-66 (R_c)	Case 58-62 (R_c)	Case 80-85 (R_a) Core 35-40 (R_c) can be secured.	Case 80-85 (R_a) Core 30-34 (R_c) can be secured.
c. Surface Temp. (T_s) °F	4	Cooling is only by oil splash/jet	Cooling is only by oil splash/jet.	Ball geometry/size not conducive to design internal cooling.	Control of cup surface temp. can be attempted by designing internal cooling.	Control of disc surface temp. can be attempted by designing internal cooling	Gear teeth surface temp. control is difficult.

A	B	C	D	E	F	G	H
d. Tip Relief (in)	4,87	Desirable range 0.0004 to 0.0006	None	Not applicable and hence configuration is not subject to this influence.	Not applicable and hence configuration is not subject to this influence.	Not applicable and hence configuration is not subject to this influence.	Reportedly difficult to control if specified 0.0004 in relief is reported to increase l.c.c. by 50%.
e. Alignment	4,87	Perfect alignment desirable.	Perfect alignment desirable.	Not difficult since ball geometries align themselves.	Possibility of edge loading exists, but can be eliminated by designing crowned cups.	Crowned discs can be designed and used thus avoiding misalignment chances.	Reportedly difficult to align gears perfectly. A misalignment by 0.005 radians is reported to decreases l.c.c. by 35%.
4. FAILURE MONITORING & DETECTION	4,87	Gears score and scuff without any warning and therefore real-time (continuous) monitoring of surfaces is desirable to detect: • friction/wear • scoring-load • post-scoring recovery time	Scuffing is monitored for 10 mts. at each load. The effect of test duration on scuffing is not clearly understood. Presently applied gear loads are increased step by step and area of tooth scuffed is estimated at each load. The 22.5% area scuff load is determined by a plot of load V_s area scuffed. This procedure is subject to errors even by closed photo-evaluation procedure. Also, accurate area estimation is not easy and subject to operator errors.	• Real-time monitoring of wear-debris is difficult for the small cup and sample size involved in 4-Ball configuration. To attempt radio tracer technique here the source of debris (balls) are too close to the oil cup where debris collects and as such possibilities of interference in the 'activity' measurements exist. • Real-time surface monitoring is difficult as contact surfaces are always hidden from operator sight. • Real-time friction/noise/vibration monitoring is feasible.	• Real-time monitoring of wear-debris can be attempted. • Real-time monitoring of cup surfaces can be attempted. • Real-time monitoring of friction/noise/vibrations can be attempted.	• Real-time monitoring of wear-debris can be attempted. • Real-time monitoring of surfaces of discs to detect first evidence of scoring can be attempted. • Real-time monitoring of friction/noise/vibrations can be attempted.	• Real-time monitoring of wear-debris can be attempted. • Real-time monitoring of surfaces to detect first evidence of scoring is difficult since there are multiple pairs of contact surfaces to monitor based on no. of pairs of teeth coming into contact. • Real-time monitoring of friction/noise/vibrations can be attempted.

A	B	C	D	E	F	G	H
5. PRECISION	1,3		Scatter of data seen presently at 95% confidence level (Ref.). Repeatability: Two determinations (A & B side of a test gear) in the same apparatus should not differ by more than 787 lbs. f/in. Two pairs of tests are acceptable by the above criterion if the averages of these pairs do not differ by more than 557 lb.f./in. Reproducibility: Single observations taken at two different installations must agree within 787 lb.f./in (95% confidence level) The averages from the two installations must agree at the 95% confidence level, within 664 lb.f./in. [Average rating of standard reference oil = 2960 lb.f./in. with average standard deviation of 284 lb.f./in].	Similar performance at 95% confidence limit established for 5 oils in a round robin programme, for load wear index and weld point at both 1760 rpm (two laboratories) & at 1440 rpm (3 laboratories) Repeatability: a. On load wear index duplicate tests should not differ by more than 17% of mean. b. In weld point, duplicate tests should not differ by more than one load increment. Reproducibility: a. In load wear index, results submitted by each of two laboratories should be considered suspect if the results differ by more than 44% of the mean. b. In weld point, results submitted by each of two laboratories should be considered suspect if the two results differ by more than one increment loading.	The following criteria should be used for judging the acceptability of results (95% confidence) Repeatability: Duplicate results by the same operator should be considered suspect if they differ by more than 22% of the mean value. Reproducibility: The results submitted by each of two laboratories should be considered suspect if the two results differ by more than 55% of the mean value.	Precision to be established by round robin tests. Precision values postulated to be better than Ryder in view of better control on: a. characterizations of results by real-time monitoring for wear and distress. b. Alignment due to use of crowned discs. c. Surface roughness of specimens after 'break-in' due to pre-finishing of discs using standard grinding wheels/silicone carbide paper. d. Specimen surface temp.Control and monitoring possibilities.	Precision to be established by round robin tests. Precision values postulated to be no better than Ryder in view of difficulties in controlling: a. alignment b. surface roughness of gears after 'break-in'. c. Surface temp. of gears. d. Operator errors in characterization of results.

slide/roll ratio to simulate gear-mesh action of aircraft gears and the Ryder Test Gears. Some of the critical parameters on which the four test configurations have been compared in this table will be explained briefly in the following paragraphs.

6.2.2.1 Type of Motion

In Section 4.1 the gear motion kinetics which assist formation of films in controformal contacts were discussed. The type of motion which can be secured in a film strength apparatus should be as close as possible to the type of motion encountered in the application of the lubricant which is being rated in the said apparatus. Most correlations fall apart if there is poor simulation in this respect. In order to simulate the type of sliding and rolling with low slide/roll ratio encountered in gears, it has been shown that a disc on disc or gear on gear is the preferred configuration.

6.2.2.2 Versatility

The test configuration should be versatile so as to obtain a variety of sliding/rolling speeds, Hertz contact stresses, desired metallurgy of contacting specimens, and desired surface finish of specimens. Also, since the use of new test specimens is costly, the selected configuration should be amenable to recycling/refurbishing of specimens if required. The four configurations compared in Table 6-17 will now be examined from the versatility standpoint.

6.2.2.2.1 Sliding/Rolling

The choice of a test configuration to replace the Ryder Test and yet simulate operational features of the aircraft gears, have been examined based on considerations such as rolling velocities across the line of action, sliding velocity (max) and slide/roll ratio. It is not necessary to simulate pitch line velocities where there is pure rolling and no sliding in gears. The worst conditions of high sliding/rolling are encountered at the gear tips which wear out, score and scuff much earlier than pitch line areas, where if any failure occurs, it is only due to fatigue (pitting). Four-ball and Timken configurations offer low sliding speeds (V_s) as indicated in Table 6-17, as compared to what is desirable to simulate Ryder Gear Test conditions. Moreover, since one of the specimens is fixed, the slide/roll ratio becomes infinite at the surfaces where distress is characterized, hence not desirable.

6.2.2.2.2 Hertz Stress

Since film strength apparatus come in a variety of shapes, it is not possible and not necessary to correlate actual contact loads between two configurations. However, the Hertz contact stresses can be simulated to give similar stress factors in static conditions. The range of Hertz stresses encountered in aircraft gears has been stated to be 125,000 to 250,000 in Reference (88). As against this requirement, the Hertz stresses encountered in four-ball and Timken machines are indicated in Table 6-17. In Table 6-18, the necessary choice of test loads and specimen curvatures have been indicated to simulate the Hertz stresses for maximum stress situations. This is important, since failures generally occur when maximum stress regimes are reached. In disc and gear configurations, this simulation is easy to achieve.

6.2.2.2.3 Metallurgy and Surface Finish

Lubricants are formulated using a variety of additive chemicals (see Section 4.0). Particularly for gear lubrication, it is important to simulate the contact metallurgy of test specimens used in film strength apparatus with that of actual applications. The protective films formed and their strength are related to the metallurgy and the additive chemical present in the oil. Also, surface roughness has some influence on the effectiveness of adsorbed as well as reacted films. Certain phosphorous films are stated to be ineffective if roughness is more than 10 micro-inch while phosphorous additives are the most commonly used film strength agents in aviation lubricants. It is therefore important to simulate test surface roughness with actual application situations in order to have a realistic assessment of performance of film strength additives, particularly phosphorous additives. The four-ball test rig is

TABLE 6-18

CALCULATION OF AVERAGE HERTZ PRESSURE

DISC MACHINE	LOAD W	DISC RADIUS				CROWN RADIUS				POISSON'S RATIO ν	YOUNG'S MODULUS OF ELASTICITY			HERTZ CONTACT AREA $A = \pi ab$	AVERAGE HERTZ PRESSURE $P_H = W/A$
		R_1	R_2	R	Hertz 1/2 width 'a'	r_1	r_2	r	Hertz 1/2 width 'b'		$E_1 \times 10^6$	$E_2 \times 10^6$	$E \times 10^6$		
	lbs	in	in	in	in	in	in	in	in	-	psi	psi	psi	in^2	psi
AFAPL	50	2	2	1	0.0108	18.666	18.666	9.3336	0.0228	0.3	30	30	16.43	0.00077	64,935
AFAPL	6200	2	2	1	0.0545	18.666	18.666	9.3336	0.1140	0.3	30	30	16.43	0.01950	317,948
MT1	50	2	2	1	0.0108	20.00	20.00	10.0000	0.0234	0.3	30	30	16.43	0.00079	63,291
MT1	4000	2	2	1	0.0468	20.00	20.00	10.0000	0.1010	0.3	30	30	16.43	0.01480	270,270
MT1	3000	2	2	1	0.0426	20.00	20.00	10.0000	0.0916	0.3	30	30	16.43	0.01230	244,698
MT1	3200	2	2	1	0.0435	20.00	20.00	10.0000	0.0936	0.3	30	30	16.43	0.01280	250,200

Note: 1. Formula used for Calculation of Hertz-Half Width

$a = 3/4(WR/E)^{1/3}$; $b = 3/4(Wr/E)^{1/3}$ . . . Reference 97, section F1 (Nature of Surfaces and Contact)

2. Formula used for Calculations of Equivalent radius

$1/R = 1/R_1 + 1/R_2$; $1/r = 1/r_1 + 1/r_2$

3. Formula used for Calculation of Equivalent Young's Modulus of Elasticity

$1/E = (1-\nu_1{}^2/E_1 + (1-\nu_2{}^2/E_2$, where ν_1 and ν_2 are Poisson's ratio.

the only alternative that has difficulty meeting required material surface constraints.

6.2.2.3 Specimen Recycling

In Table 6-17, we have shown that disc specimens and Timken test block and cup specimens can be recycled while gear and ball specimens cannot by recycled. Machining/grinding of discs in easy but gears once distressed (scuffed) cannot be reground without loosing alignment and dimensional accuracies. With discs, even if diameters have to be reduced in the recycling process, desired sliding/rolling speeds can be secured by slightly varying the test speeds. This is a desirable feature to have, since the cost of test specimens is a significant test cost factor.

6.2.2.4 Control on Operational Parameters

The choice of a proper test configuration to replace the Ryder Gear Test also depends on the extent of control a configuration can provide on parameters directly influencing test precision. Since the present study is directed towards securing better precision, any hardware se= lected should have the least influence on parameters such as alignment and tip relief which are considered the most critical parameters on which control cannot be exercised in the Ryder Gear Test. In this respect, disc on disc and ball on ball configurations are the best since they align themselves easily and there is no tip relief influence. Disc configuration is also amenable to the control of surface roughness in the break-in process and surface cooling during the testing process.

The ball on ball configuration (four-ball) is not good from the point of view of amenability to cooling of specimens during test, since design of internal cooling of small ball geometries is difficult to achieve. In gears, the gear tips are far away from the core and therefore internal cooling design through the core is difficult. A good deal of experimentation with alternative procedures would be required to secure a good hold on specimen cooling and uniformity in specimen roughness after break-in of the specimens, but it is feasible with disc on disc configuration.

6.2.2.5 Failure Monitoring

The alternative test hardware and test configuration selected should be amenable to real time monitoring of wear debris, friction trace, and rubbing surface in order to incorporate the new characterization criteria outlined in Section 5.3. In this respect, the disc configuration is the best choice and techniques can be developed for single contact surfaces of disc machines as compared to multiple contact surfaces obtained in gear machines. In configurations like the four-ball and Timken, real time surface monitoring of the specimen is not possible as the distressed surface is always hidden from operator sight during progression of the test. Monitoring is feasible with the disc on disc configuration since an observation can be maintained constantly on the surfaces entering and leaving the contact conjunction. With the gear on gear configuration, monitoring is difficult to accomplish since all gear pairs coming into mesh have to be watched as scoring (initial) can first appear on any of the gear pairs. Real time monitoring of the friction trace is easy with disc on disc, ball on ball, and Timken configurations but it is difficult with gear on gear configurations. Wear debris monitoring is similarly difficult with the four-ball configuration but feasible with the others.

6.2.3 Configuration Selection Summary

Based on the above configuration selection criteria, the disc on disc approach is the best alternative for load-carrying capacity testing.

6.3 Review of Existing Disc Machines

In Table 6-19, some of the disc machines being developed or sighted in literature have been tabulated. They have been examined from the point of view of suitability to adopt the new three step characterization criteria described in Section 5.3 and versatility to secure the speed, load, and metallurgical simulations desired.

TABLE 6-19

REVIEW OF EXISTING DISC MACHINES TO TRY NEW CHARACTERIZATION CRITERIA (3 STEP)

TEST HARDWARE	REF	FRICTION TRACE REALTIME MONITORING	WEAR RATE REAL TIME MONITORING	INITIAL SCORING REALTIME MONITORING	POST SCORING RECOVERY REALTIME MONITORING	WHETHER (3),(4),(5), (6) CAN BE ATTEMPTED	VERSATILITY (LOADS, SPEEDS, METALLURGY ETC.)	SPECIMEN TEMP. CONTROL	WHETHER CROWN DISCS USED/CAN BE USED FOR MINIMIZING ALIGNMENT ERRORS
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
AFAPL Disc tester (USA)	87 89	Attempted through torque monitoring	Not attempted	Not attempted	Not attempted	YES - (3) NO - (4),(5),(6)	FAIR	Specimen temp. measurement attempted but not controlled.	YES
CATEPILLAR Disc tester (USA)	87		Not attempted	Not attempted	Not attempted	(3) - NO - (4),(5),(6)	FAIR		YES
AMSLER MACHINE (Switzerland)	93	Attempted	Not attempted	Not attempted	Not attempted	YES - (3) NO - (4),(5),(6)	POOR Speeds (200 to 400 r.p.m.	Not attempted	Can be used
MOBIL CORYTON RIG (UK)	91		Not attempted	Not attempted	Not attempted	(3) - NO - (4),(5),(6)	FAIR Speeds 50 to 5000 r.p.m.	Attempted	Can be used
GEAR ROLLER TESTER (Germany)	94		Not attempted	Not attempted	Not attempted	(3) - NO - (4),(5),(6)			Can be used
EDGE TO EDGE DISC M/C (UK)	46	Attempted.	Not attempted	Not attempted	Not attempted	YES - (3) NO - (4),(5),(6)	POOR Speeds 72 to 700 r.p.m.	Temp. monitoring attempted but not control	Can be used

6.3.1 Characterization Criteria

The new characterization criteria has not as yet been attempted in any of the existing machines listed. None of the machines listed however, in this table have reached a stage where they can be adopted or used.

6.3.2 Versatility

The speed and load ranges of the AFAPL Disc Tester, the Catepiller Disc Tester and the Mobil Coryton Rig are fair from the point of view of achieving total desired simulation with respect to sliding speeds, and slide/roll ratio. However, it has not been possible in this study to go into the details of respective drive systems in order to assess total rig versatility.

The continuous loading system required to adopt three step characterization criteria described in Section 5.3 cannot be secured in any of the existing disc machines.

Sufficient information required to make a judgement on disc cooling adaptability has not been totally collected.

6.3.3 Alignment

All disc machines considered can use crowned discs in order to secure alignment of contact surfaces.

6.3.4 Disc Machine Summary

Based on the above analysis, no existing disc test rig is suitable for utilization and/or modification as a cost effective load-carrying capacity discriminator.

6.4 Proposed Advanced Disc Machine

Based on the above nonsuitability assessment of existing disc test rigs, the sole remaining alternative is the development of a new advanced disc machine for gear lubricant load-carrying capacity determination. This machine will incorporate design parameters required to accommodate the hardware and criteria modifications/constraints necessary for efficient simulation and high precision.

Detailed design specifications are listed in Table 6-20. Highlighted specifications include disc temperature cooling, continuous loading, real time monitoring of disc wear rate, and surface condition interface friction.

6.5 Advanced AFAPL Disc Machine Comparison

As a result of the existence of the current AFAPL Disc Tester, a comparison will be highlighted between this rig and the proposed "advanced" disc rig.

Table 6-20 compares the conceptual design features of the proposed advanced disc machine with the AFAPL disc tester. The improved simulation in Hertz stresses, sliding and rolling speeds and also slide/roll ratio are indicated. Table 6-18 gives the calculations made for simulating Hertz stresses and compares those obtained with the AFAPL disc machine. Appendix D gives the calculations made to obtain the sliding and rolling velocities and also slide/roll ratio presently encountered in the Ryder Gear Test.

6.5.1 Sliding/Rolling Speeds and Ratio

The proposed disc machine will closely simulate the sliding/rolling speeds and ratio exhibited by the Ryder rig. This consistency will serve to enhance qualification testing, in particular, the testing of Ryder qualified lubricants. AFAPL rig sliding/rolling speeds and ratio do not closely simulate Ryder characteristics.

TABLE 6-20. COMPARISON: AFAPL DISC TESTER AND PROPOSED ADVANCED DISC TESTER

A	B	C	D	E	F	G
CRITICAL PARAMETER	REF	Desired goal for simulation with Aircraft Type gears	Simulation with Ryder test gears	AFAPL DisC Tester	Proposed Advanced Disc Tester	Remarks
1. TYPE OF MOTION	5,87	Sliding and rolling as in any spur, helical or bevel gear, operating under low slide/roll ratio.	Sliding and rolling as in any spur, helical or bevel gear, operating under low slide/roll ratio (see item 2).	Discs are driven independently in opposite directions of motion. Sliding velocity (V_s) to be the difference of surface speeds V_1 and V_2 with low slide/roll ratio (see item 2).	Conceptually no change proposed in type of motion. Sliding speeds and slide/roll ratio to be better simulated with those of Ryder-test. (See item 2).	
2. <u>VERSATILITY</u> a. Gear/Disc Dia and Geometry	88,1,87 Tab. 18	Gears of 2 to 8" PCD.	Gears of 3.5 inch PCD, $22^1/_2{}^o$ pressure angle with 28 teeth per gear, 8DP.	Discs of 4 inch diameter with crowns of radius 18.666 inch.	Discs of 4 inch diameter with crowns of radius 20.000 inch, for better simulation of hertz-stress in aircraft gears.	
b. Gear/Disc Thickness	1,87	-	Wide gear - 0.937 inch Narrow gear - 0.250 inch	Thickness - 0.666 Inch	Thickness proposed to be 1 inch.	
c. Rotation	1,87 Tab. 18	-	10,000 r.p.m.	Disc speed range 716 to 3342 rpm	Disc speed range proposed to be 2100-4600 rpm, to better simulate rolling velocity, sliding velocity and slide/roll ratio of ryder gears.	
d. Pitch-Line Rolling Velocity (V_p)	1	-	1,833.33 ips	-	-	
e. Rolling Velocities Across Line of Action High (V_1) Low (V_2)	96		963.39 ips 439.79 ips	700 ips 150 ips	963.80 ips 440.80 ips	
f. Sliding Velocity (V_s) (Max)	Ap-D		523.60 ips	550 ips	523.80 ips	
g. Slide/Roll Ratio (Max)	Ap-D		0.373	0.647	0.373	

A	B	C	D	E	F	G
h. Hertz Stresses (psi)	76 Tab. 18	125,000 - 250,000	-	64,934 to 317,948 for load range 50 to 6200 lbs.	63,291 to 250,200 for load range 50 to 3200 lbs. (better control in higher load range of 2000 to 3200 lbs. where failures likely to occur.	
i. Metallurgy of Specimeos	88,1,87 Ap. E	AMS 6260, AMS 6265	AMS 6260	AMS 6260	AMS 6260	
j. Surface Roughness of of Specimens • Trans-verse • Axial • Compo-site (siogle disk/ gear)	4, 87,89		Desired 9 to 14.5 μin.AA	8-10 μin.CLA 16-20 μin.CLA 12-15 μin.AA	To be decided. Discs are proposed to be axially ground to simulate with machining marks in gears and finished with suitable grinding wheel/ silicon carbide paper, axially. Prior to qualificatioo testing, each disc pair is proposed to be run-in by a standard procedure to be developed to control composite roughness of a single disc to 8-10 inch.	
k. Surface Tex-ture of Cears/ Specimens	74,75	Axially ground	Axially ground	Honed/circumferentially ground	Axially ground	
l. Heat Treatmeot of Specimens/ Cears and Hard-ness	37,89 4	Carburized	Carburized Case hardness: 81-84(R_a) Core hardness: 35-40(R_c)	Carburized (experiments also done with nitrided discs.)	Carburized, case and core hardness to be decided.	
m. Specimen Temp. Control	37,89	None	Desirable	Varied from oil temp. of 400°F to maxdisc temp. of 600°F	It is proposed to control the quasi steady state disc temps. in the range 375 to 425°F by interoal cooling of discs and use of hollow discs.	
n. Oil Jet Temp.	4,87, 89	-	165°F	Results reported at temps 140, 190, and 275F	To be decided	
o. Lubricants Tested		MIL-L-7808, 23699 (Current Revisions)	MIL-L-7808,23699 (Current Revisions)	MIL-L-7808, MIL-L-23699 and other gear/E.P. lubricants.	Proposed to be used for both MIL-L-7808 and MIL-L-23699. (Current Revisions and other gear/E.P. lubricants.	

A	B	C	D	E	F	G
p. Type of Lubrication	87,89	EHD & Boundary	EHD & Boundary	EHD & Boundary	EHD & Boundary	
3. TEST DURATION	1,87, 89		10 mins at each load (increased at 50 lb. increments) plus stopping and starting time plus time for inspection of gears and rating of area scuffed. Process to be repeated till $22^1/_2$% of area is scuffed.	3 mins at each load increased at 50 lb. increments. Process to be repeated till scoring load is reached.	To be less than that of both AFAPL tester and 'Ryder' in view of proposed continuous loading system (rate of increase to be decided) and real-time monitoring of surfaces for wear and surface distress.	Advanced disc test technique to be developed by experimentation and research.
4. QUALIFICATION COSTS*	see note		1980 - Air Force estimates $3,300 1980 - ALCOR Catalog: $2,252		Estimates: $1,080 (without recycling of discs) $600 (with recycling of discs)	Advanced disc test technique to be developed by experimentation and research.
5. HARDWARE COSTS	see note		$300,000 **		Estimates: For Prototype: $150,000 - 200,000	Advanced disc machine to be designed, fabricated and tested.
6. CORRELATION WITH FIELD (known/ postulated)	-	Exact			Postulated to be good due to more precise simulation with actual aircraft gears in motion, metallurgy and slide/roll considerations.	Advanced disc machine to be designed, fabricated and tested.
7. PRECISION	2,3		Poor		Postulated to be good due to better characterization techniques and better control on perceived parameters affecting precision.	

Note: * Due to lack of precision eight determinations have to be presently made for qualification to MIL-L-7808G, Amendment 2, 9/10/71, with Ryder Gear Test. In the case of advanced disc because of improved precision, three determinations are sufficient to quantify L.C.C. performance.

** Approximate ball-park figure obtained from ERDCO, Engg, Inc., Adison Illinois, by private communication of 7/13/81.

A	B	C	D	E	F	G
8. FAILURE MODE AND DETECTION	1,87, 89	Detection of the following types of lubrication related surface distress modes and estimating the performance of the lubricant on the basis of the intensity of the same: (a) Wear (b) Scoring/Scuffing	Estimation of load carrying capacity of lubricant based on scuffing load at $22^1/_2$% area scuffing of gears by running the gears at loads higher than scoring loads. (criteria is questionable since the post-scoring operation of the specimens is more a test for material resistance property under post-seizure conditions rather than a test for lubrication capacity of the lubricant under test)	(a) No attempt made to quantify actual wear volume/wt. loss of test discs. (b) Scuffing load characterized by the minimum applied load at which complete break-down of film-contact-resistance occurs with visual evidence of disc scuffing which is assessed after stopping the test.	(a) Wear of specimens to be quantified by real-time monitoring techniques to be selected. (b) Loading is proposed to be made a continuous variable with time at a regulated rate. (c) Scoring-Load to be instantaneously detected both by real time friction trace and by constant watch on the contact surfaces. (d) 'Post-Scoring-Recovery Time' to be detected by the real-time needed to reach Steady-State Conditions in friction/vibrations/noise, after the 'on-set' of scoring. Where no such recovery is observed in a reasonable length of time (to be specified) the lubricant is said to have no capacity to recover from scoring.	Advanced disc test technique to be developed. Advanced disc test technique to be developed. Advanced disc test technique to be developed. Advanced disc test technique to be developed.

6.5.2 Hertz Stresses

The advanced disc machine will provide closer simulation of maximum Hertz stresses encountered in aircraft gears than the AFAPL disc tester. This is important since failure generally occurs when maximum stress regimes are reached.

6.5.3 Surface Roughness

The surface roughness (composite) for a pair of discs in the AFAPL tester is in the region of 12-15μin., while most phosphorous additives used in oil formulations reveal their effectiveness if roughness is 10μ in. or better. The advanced disc machine will use disc finishes in the 8-10 μin. range. This is proposed to be achieved by developing a suitable running-in procedure.

6.5.4 Surface Texture

The disc for the advanced machine will be ground axially to simulate grinding marks of gears. This is not considered with AFAPL discs.

6.5.5 Specimen Temperature Control

The advanced disc machine will be designed in order to accommodate the cooling of discs and control of disc surface temperature between driving test. No such capability has been built into AFAPL disc machine.

6.5.6 Test Duration

In the AFAPL test procedure, three minutes are spent at each test load starting from 50 lbs. Thus, nearly 1 hour to $1^1/_2$ hours is required to reach failure loads for MIL-L-7808 type oils. The continuous loading system of the advanced disc machine could reduce testing time to between $^1/_2$ hour to 1 hour.

6.5.7 Precision

The precision aspects of the present AFAPL tester could not be ascertained in the present study. With real time monitoring capabilities, and better control on perceived test parameters, the precision of the advanced disc machine is postulated to be much better than Ryder Test.

6.5.8 Advanced/AFAPL Disc Machine Comparison Summary

Based on the above analysis, the proposed advanced disc machine exhibits clearcut advantages over the present AFAPL machine design. These advantages will enhance machine simulation efficiency and precision as well as minimize costs.

6.6 Cost Considerations

Figure 6-3 gives a comparative idea of costs between the present Ryder Test and the proposed advanced disc machine test. This has been indicated for the test hardware as well as for cost per sample of lubricant tested. An estimate of the hardware cost of the Ryder gear rig was obtained from ERDCO Engineering Corporation, Evanston, Illinois. The estimates for the proposed advanced disc machine are for the first prototype. Subsequent machines are likely to cost less (≃ 20% less), since the amount spent in initial designs/drawings for the first prototype can be saved for subsequent machines.

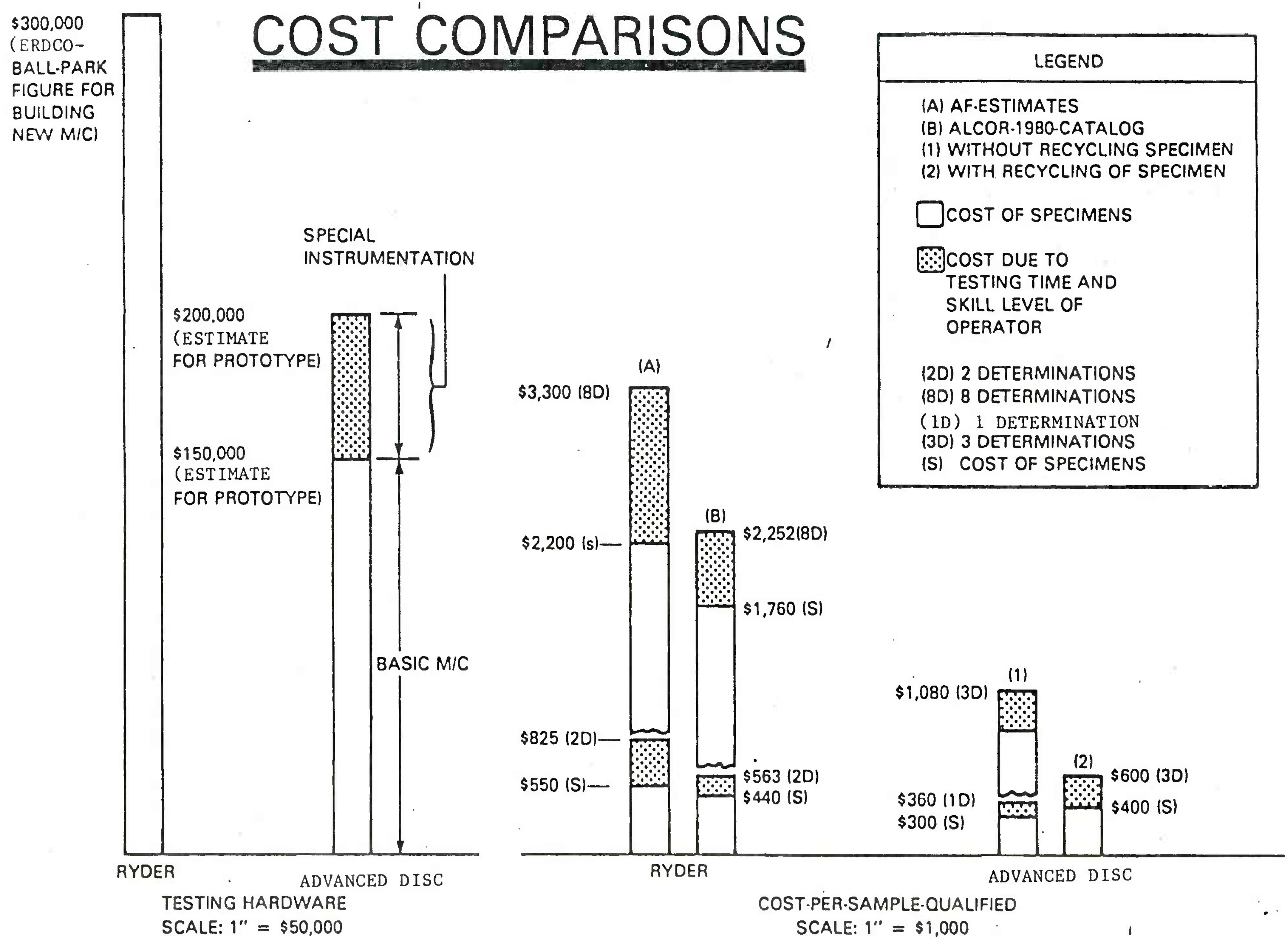
In calculating the present Ryder test costs, the costs for two determinations (one pair of test gears) as well as for eight determinations (four pairs of test gears) have been indicated. The eight determinations are the required number of determinations prescribed in MIL-L-7808G, Amendment 2 in view of the precision problems presently encountered.

With the improved precision predicted for the advanced disc machine test it is expected that a maximum of only three determinations would be required for qualification of MIL-L-7808 type oils.

As indicated in Figure 6-3, the proposed advanced disc machine is estimated to cost between $150,000 to $200,000 for the first prototype. With subsequent machines, the hardware costs are likely to come down by 5% to 20%. This may be compared with $300,000 quoted as the present cost of a new Ryder gear machine.

The conservative estimate of qualification test cost (eight determinations) with the Ryder Gear Test is $2,252 for one candidate oil sample. The maximum advanced qualification test cost is estimated at $1,080 (three determinations) due to expected improved precision and savings in test time. The qualification test cost with the advanced disc machine could decrease to about $600 per candidate oil sample if recycling/refurbishing of the disc test specimens is implemented.

FIGURE 6-3


COST COMPARISONS
LEGEND
(A) AF-ESTIMATES
(B) ALCOR-1980-CATALOG
(1) WITHOUT RECYCLING SPECIMEN
(2) WITH RECYCLING OF SPECIMEN
COST OF SPECIMENS
COST DUE TO TESTING TIME AND SKILL LEVEL OF OPERATOR
(2D) 2 DETERMINATIONS
(8D) 8 DETERMINATIONS
(1D) 1 DETERMINATION
(3D) 3 DETERMINATIONS
(S) COST OF SPECIMENS
$300,000 (ERDCO-BALL-PARK FIGURE FOR BUILDING NEW M/C)
SPECIAL INSTRUMENTATION
$200,000 (ESTIMATE FOR PROTOTYPE)
$150,000 (ESTIMATE FOR PROTOTYPE)
BASIC M/C
RYDER
ADVANCED DISC
TESTING HARDWARE
SCALE: 1" = $50,000
(A)
$3,300 (8D)
$2,200 (s)
$825 (2D)
$550 (S)
(B)
$2,252(8D)
$1,760 (S)
$563 (2D)
$440 (S)
(1)
$1,080 (3D)
$360 (1D)
$300 (S)
(2)
$600 (3D)
$400 (S)
RYDER
ADVANCED DISC
COST-PER-SAMPLE-QUALIFIED
SCALE: 1" = $1,000

7.0 CONCLUSIONS

Based on the investigations and subsequent analyses conducted under this program, the following conclusions have been formulated.

1. Load-carrying capacity of aircraft engine oils; MIL-L-7808, and MIL-L-23699, is a perceived property and not an inherent property. Physico chemical tests cannot predict this film strength property, therefore, finite component/system tests are required for qualification.

2. There currently exists a firm requirement for a load-carrying capacity assessment technique.

3. The ASTM D-1947 test, commonly referred to as the Ryder Gear Test, lacks precision and is losing the confidence of its users. Its utilization is in turn decreasing. Due to lack of precision as many as eight determinations have been specified in specifications like MIL-L-7808G, Amendment 2 (9/10/71) thereby increasing qualification costs for load-carrying capacity.

4. The shortcomings of the Ryder Gear Test are as follows:

(a) Arbitrary load-carrying capacity (L.C.C.) rating procedure based on the average and estimated value of 22.5% scuffing load whose measurement is subject to operator errors in both open operator estimation system as well as closed photo-evaluation system.

(b) In addition to errors of determination, the 22.5% scuff load criteria assumes a build-up of scuff related distress with time at each load and test duration. This assumption cannot be supported by theoretical thin film lubrication concepts.

(c) Heavy influence of test gear variables such as tip relief, profile roughness, hardness, and alignment which cannot be precisely controlled from test to test, thus affecting precision.

(d) Heavy influence of operational parameters such as load, speed, and specimen temperature which cannot be precisely controlled from test to test, thus affecting precision.

(e) Influence of the differences in the construction of the three test heads permitted for the ASTM D-1947 test, which adds to lack of test precison.

4. Modification of the Ryder Gear Test is not cost effective.

5. No alternative testing techniques presently exist that accurately and precisely determine load-carrying capacity.

6. The major shortcomings of alternative existing film strength test rigs are as follows:

(a) Inability to simulate gear meshing action,

(b) Inability to simulate relevant sliding/rolling speeds and ratios,

(c) Inability to simulate pertinent stress factors,

(d) Unacceptable test specimen metallurgy,

(e) Limited versatility

(f) Poor precision,

(g) Unacceptable rating criteria,

(h) Unacceptable control of operational parameters,

(i) Specimen temperature uncontrolled,

(j) Specimen variation.

7. A new relevant test criteria approach is required. This approach consists of real time monitoring of wear, scoring load, and post scoring load recovery. Criteria relies on continuous loading of test specimens.

8. Test configuration analysis has identified disc on disc as the optimal approach to load-carrying capacity determination. This selection is based on the following factors:

(a) Good simulation efficiency with respect to Hertz stress, gear mesh action, sliding and rolling speeds, low slide roll ratio.

(b) Suitable specimen metallurgical, surface roughness and texture.

(c) Adaptability to specimen cooling/temperature control.

(d) Adaptability to new three step criteria and respective real time monitoring requirement.

(e) Adaptability to precision control of specimen quality.

(f) Adaptability to precision control of operation parameters and respective parameter range requirements.

9. Existing disc on disc test machines would require major modifications in order to suit the new precision design specifications.

10. A new high precision advanced disc machine could be developed based on the above outlined hardware and test criteria design specification.

11. The proposed advanced disc machine incorporates the critical features identified in this study, in order to improve precision in load-carrying capacity determinations of lubricants.

12. The proposed advanced disc machine should lower qualificaton costs for load-carrying capacity determinations by at least 1/2 (per sample), on the basis that (3) tests are sufficient per characterization, due to expected better precision proposed to be achieved.

13. The new advanced disc machine can be fabricated at about 2/3 the cost of a present Ryder machine.

8.0 RECOMMENDATIONS

Based on the conclusions presented in Section 7.0, the following recommendations are presented.

8.1 Near Term

1. Design and fabricate the proposed advanced disc machine.

2. Develop the new three step test criteria/characterization technique for determining load-carrying capacity by real time monitoring of wear and surface distress.

3. Upon advanced disc machine fabrication, conduct a rigorous test program in order to verify machine specifications, quantify precision, develop test criteria, and develop a correlation with the Ryder test.

8.2 Long Term

1. Fabricate several advanced disc test machines, or proposed lines.

2. Conduct testing to assess reproducibility.

3. Introduce advanced disc machine into pertinent lubrication specifications.

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APPENDIX

APPENDIX A

GEAR MACHINES - DATA

GEAR TEST MACHINES	PINION SPEED RPM	TEST OIL FLOW	TEST OIL TEMP °C	LOAD SEQUENCE - STEPS	FAILURE CRITERIA	TIME AT LOAD MIN	DISTANCE BETWEEN CENTER LINES	LUBE SYSTEM	NUMBER OF TEETH	PITCH MM	WIDTH OF GEAR MM	MAX LOAD AT CONTACT Kgf/cm^2	DIAMETRAL PITCH
IAE (IP-166)	6K MAX	*V	110	5 lb.	60% SCUFF	5	82.6	JET	16	5.0	4.8	35.6K	4.8
RYDER (STD) (ASTM D-1947)	10K MAX	*V	74	5 lb.	22.5% SCUFF	10	88.9	JET	24	6.4	6.4	20.5K	8
FZG (DIN 51354)	4.4K MAX	*V	90	20 kg NET	Wt. LOSS & Wear Rate	15	91.6	DIP	28	20.1	20.1	20K	-
FOUR-SQUARE TEST M/C (GLEASON)	SIMILAR IN PRINCIPAL TO RYDER, CONTINUOUS AND SHOCK LOADS USED												
NASA LRC GEAR FATIGUE	10K				VIB			JET	28				8

* V = Variable

APPENDIX B

TEST GEARS - DATA

TEST GEARS	NO. OF TEETH	PRESSURE ANGLE (°)	TIP RELIEF (IN)	PITCH DIAMETER MM (IN)	CENTER DISTANCE (IN)	TOOTH WIDTH MM (IN)
WADD	28/28	22.5	4×10^{-4}	3.5	3.5	
WADD (EXP)	28/28	19	25×10^{-4}	3.5	3.5	
RYDER	28/28	22.5	0	3.5	3.5	2.5
FZG A	16/24	20				20 MM
FZG C	16/24	20				20 MM
FZG L	12/16	20				20 MM
NASA	28/28			3.5		
IAE	15/16	26.3	5×10^{-4}			.188

APPENDIX C

DISC MACHINES - DATA

DISC TEST MACHINES	SPEED RANGE, RPM	LOADING	FAILURE CRITERIA	DISC WIDTH (MM) IN	TEMP °C (OIL)	LOAD STEPS	CROWN RADIUS (IN)	TEST OIL SYSTEM	MAXIMUM LOAD	DISC MATERIAL
MOBIL CORYTON RIG	5000 MAX	HYD	TORQ & TEMP			*V				
AMSLER DISC MACHINE	200-400	SPRG.	WEAR & SCUFF	10		*V			200 Kg	EN 31
EDGE TO EDGE	700 MAX	MECH.	TORQ	8.9		*V				EN 58B/ EN 31
AFAPL DISC TESTER	5000 MAX	HYD	SCORE/ CONTACT RESISTANCE	4 IN.	400°F MAX	*V	18.66	5 Gal.	6.200 LBS.	AMS 6260/ 6265
CATERPILLAR DISC M/C				3 IN.			14 IN.			
GEAR ROLLER TESTER		POWER	FATIGUE			*V				42 Cr. $M_o{}^4$

*V = Variable

APPENDIX D

Sliding and Rolling Speed Calculations

With reference to figure 2, triangles CDE_1 and O_1AC are identical, also triangles CDE_2 and O_2BC are identical. Angle DCE_1 = Angle $AO_1C = \alpha + \beta$.

$$\overline{U_1}/\overline{O_1C} = \overline{DE_1}/\overline{AC} = \overline{CD}/\overline{O_1A} \qquad 1$$

or

$$\overline{U_1}/\overline{O_1C} = \overline{DE_1}/r_1 = \overline{CD}/\overline{O_1A} \qquad 2$$

But CD is the velocity of a contact point along the line of action, which is the same as the tangential velocity of a point on the base circle, which is given by:

$$\overline{CD} = w_1\overline{O_1A} \qquad 3$$

and

$$\overline{CD} = w_2\overline{O_2B} \qquad 4$$

Note: Components of velocity of both driver and driven gears along line of action (along common tangent AB to base circles) must be identical and equal to CD in order that the two involutes remain in contact.

$$\overline{U_1}/\overline{O_1C} = \overline{DE_1}/r_1 = w_1\overline{O_1A}/\overline{O_1A} = w_1 \qquad 5$$

Therefore, $\overline{DE_1} = w_1r_1$ 6

By similar logic we can show that,

$$\overline{DE_2} = w_2r_2 \qquad 7$$

Therefore, the components of velocity normal to the line of action of the surfaces (across line of action) are identical to the surface speeds of discs having the same angular velocities as the gears and having the same radii of curvature, respectively, as the teeth have at the point of contact.

The sliding velocity at contact point 'C' is equal to the difference in the velocities normal to the line of action between the surfaces and is given by:

$$V_s = V_1 - V_2 \qquad 8$$

where, $V_1 = w_1 r_1$ 9

and $V_2 = w_2 r_2$ 10

Now consider figures 1 and 2,

$O_1A = O_2B$ = Base circle Radius (B.C.R.) = 1.6168 Inch

O_1C = Outside Radius, at gear tip (O.R.) = 1.8600 Inch

Therefore, $\cos(\alpha+\beta) = O_1B/O_1C = 1.6168/1.8600 = \underline{0.8692}$

Therefore, Angle $\alpha + \beta = 29^0 - 38'$ 11

But Angle α = pressure angle = $22^0 - 30'$

Therefore, angle $\beta = 7^0 - 8'$ 12

$$r_1 = O_1C \sin(\alpha+\beta) = 1.86 \sin(29^0 - 38')$$

$$\underline{r_1} = \underline{(1.86)(0.4964)} = \underline{0.9196 \text{ Inch}} \qquad 13$$

Now, $r_1 + r_2 = AP + PB = AB$ 14

$$AP = PB = O_1P \sin\alpha = O_2P \sin\alpha$$

$$AP = PB = (1.75)(\sin 22^0 - 15')$$

$$AP = PB = (1.75)(0.3827)$$

$$AP = PB = 0.6697 \text{ Inch}$$

Therefore, $\underline{AB = AP + PB = 1.3394}$ 15

Therefore, $r_2 = AB - r_1 = 1.3394 - 0.9196$ 16

$$r_2 = 0.4198 \qquad 17$$

$$V_1 = w_1 r_1 = 2\pi(10{,}000)(0.9196)/60 \qquad 18$$

$$\underline{V_1} = \underline{963.39 \text{ ips}} \qquad 19$$

$$V_2 = w_2 r_2 = 2\pi(10{,}000)(0.4198)/60 \qquad 20$$

$$\underline{V_2} = \underline{439.79 \text{ ips}} \qquad 21$$

$$\underline{V_s} = \underline{V_1 - V_2} = 963.39 - 439.79 = \underline{523.60} \text{ ips} \qquad 22$$

$$\text{Slide/Roll Ratio (Max)} = \frac{V_s}{V_1 + V_2} = \frac{523.60}{963.39 + 439.79} = 0.373 \qquad 23$$

APPENDIX E

Bulk Properties of Steels used for Specimens (from Ref. 4)

Property	Symbol	Unit	AMS 6260** AMS 6265	AMS 6475	Remarks
Young's Modulus	E	psi	30×10^6	30×10^6	
Poisson's Ratio	v	-	0.30	0.30	
Equivalent Young's Modulus	*E	psi	33×10^6	33×10^6	* $E = E/(1-v^2)$
Density	ρ	lb/in^3	0.283	0.283	
Specific heat	c	in/F	1075	1075	
Thermal Conductivity	k	lb/F-sec	5.84	6.48	
Blok's Thermal Coefficient	β	$lb/F\text{-}in\text{-}sec^{1/2}$	42.15	44.40	$\beta = \sqrt{\rho c k}$

METALLURGICAL ANALYSES OF AMS 6260 STEEL*

	Percent of Element							
	C	Mn	P/S	Si	Ni	Cr	Mo	Al
AMS 6260F*	0.07-0.13	0.40-0.70	0.025 max	0.20-0.35	3.0-3.5	1.0-1.4	0.08-0.15	-
Sample 1	0.22	0.65	0.010	0.29	3.42	1.30	0.14	Not checked
Sample 2	0.19	0.70	0.010	0.28	3.14	1.37	0.16	Not checked
Sample 3	0.11	0.69	0.005(P) 0.007(S)	0.27	3.09	1.38	N.C.	0.029

* Data from ref. 87.
** AMS 6260 Steel is equivalent to SAE 9310 Steel/AISI E9310 Steel.

APPENDIX F

PROPERTIES OF REFERENCE FLUIDS*

	HEROCOLUBE A [a]	REFERENCE OIL C [b]
Viscosity, cs		
100F	20	237-245
210F	4.3	19
Pour Point, F	-85	10
Flash Point, F	465	470
Specific Gravity 60/60F	1.002	
Saponification Nr	420	0.5
Viscosity Index		95

* Taken from Table III-4 of Reference (2)

(a) Hercolube A is a registered trademark of Hercules, Inc. Data presented has been extracted from Hercules, Inc., Technical Bulletin S159.

(b) Data presented has been extracted from MIL-L-6082C.